AD-A032 867 COAST GUARD RESEARCH AND DEVELOPMENT CENTER GROTON CONN F/G 13/10
FULL SCALE SHIP'S HULL EXPOSURE FIRE TESTS.(U)
JUN 76 R C RICHARDS, R G VORTHMAN

UNCLASSIFIED CGR/DC-14/76 USCG-D-97-76 NL

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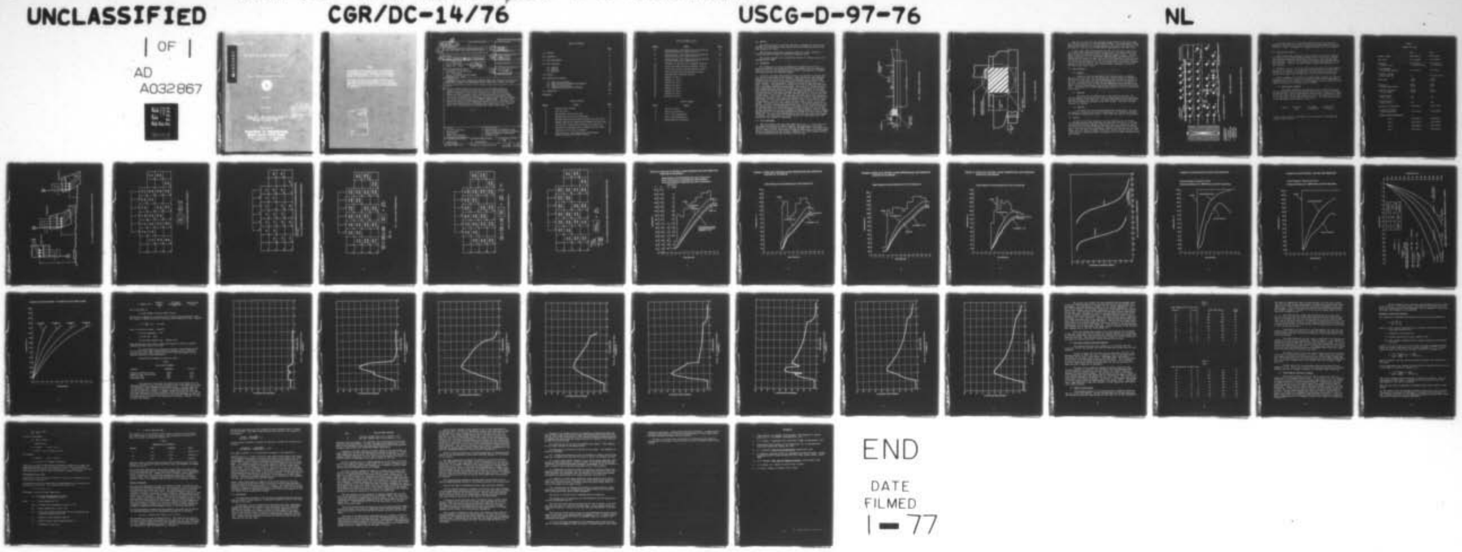

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Robert C. Richards and Robert G. Wertman

June 1976

Final Report

Prepared for

DEPARTMENT OF TRANSPORTATION

NOTICE

Technical Report Documentation Page

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
18 19 USCG-D-97-76		

4. Title and Subtitle

6 FULL SCALE SHIP'S HULL EXPOSURE FIRE TESTS.

5. Report Date

11 JUN[redacted]76

6. Performing Organization Code

7. Author(s)

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8. Performing Organization Report No.

14 CGR/DC-14/76

9. Performing Organization Name and Address

U. S. Coast Guard
Fire & Safety Test Facility
Mobile, AL 36615

12 46p.

10. Work Unit No. (TRAIS)

11. Contract or Grant No.

12. Sponsoring Agency Name and Address

Department of Transportation
U. S. Coast Guard
Office of Research and Development
Washington, DC 20590

13. Type of Report and Period Covered

9 Final Report.

14. Sponsoring Agency Code

15. Supplementary Notes

Performed at the Fire & Safety Test Facility under the technical control of the Coast Guard Research and Development Center, Avery Point, Groton, CT 06340.

16. Abstract

Fourteen hundred square feet of the hull of the MV RHODE ISLAND was exposed to two JP-5 fires located close to the aft starboard side in order to determine the rate of heat transfer to a steel hulled vessel. During the first fire test a water spray system producing an application rate of 0.12 gal/min/ft^2 was employed. The tests were analyzed to predict the results of similar fires on an aluminum hulled vessel. The heat transfer rate to the steel hull ranged from 35000 to 45000 BTU/HR. The water spray limited the initial rate of temperature buildup but its overall effect could not be quantified because of non-uniform water application. The predicted effects of a similar fire on an aluminum hull would be disastrous.

sq ft

17. Key Words	18. Distribution Statement
hull fire resistance marine safety ships fires steel fire resistance aluminum fire resistance	This document is available to the public through the National Technical Information Service, Springfield, Virginia 22161

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
UNCLASSIFIED	UNCLASSIFIED	49	

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1.0 PURPOSE

The primary purpose of this fire test was to determine the rate of heat transfer to a steel hulled vessel from an oil fire situated on the water surface alongside the vessel.

The secondary purpose was to gauge the effect of a water curtain in blanketting the hull surface of the vessel during the fire.

The tertiary purpose was to predict the results of a similar fire on an aluminum hulled vessel.

2.0 BACKGROUND

The findings of an earlier investigation(1) mandated via law that a vessel must provide fire protection for its passengers, crew, and cargo passively by virtue of its construction and actively, if necessary, by installed fire protection and fire fighting systems.

Currently, the Coast Guard requires steel as the primary structural material for the construction of tank vessels. This provides two very important properties for fire protection; steel is incombustible and it shows a high strength retention factor in the presence of elevated temperatures.(2) Also, steel in ships - unlike buildings - retains its integrity when subjected to fire. Recently, the possibility of permitting the use of materials other than steel for primary structure has been posed. Any material so allowed should be both incombustible and capable of withstanding a certain degree of fire exposure without detrimental reduction in the structural integrity of the vessel. This test series was designed to measure the temperatures to which a hull might be subjected from a fire alongside, both passively and with an active fire protection system - water spray. Certain types of vessels which carry hazardous cargoes are required to have water spray systems as a means of reducing the net heat input to the hull from exposure to exterior fires. Water spray can serve a dual purpose. Not only can it reduce the heat input to cargo, it can also provide some protection to the primary structure. Water spray systems are required to provide a minimum water application rate of 0.25 gal/min/ft^2 and thus the pumping capacity is proportional to the exposed surface area of the ship. If this application rate could be reduced and still provide an effective water curtain, then both the cost and the weight of pumping apparatus could be reduced. One definition(3) of an effective water curtain allows a net heat input of 6000 BTU/hr/ft^2. The first hull exposure test in this fire test series was designed to evaluate the effectiveness of a water curtain with approximately half the application rate required by regulation.

3.0 TEST ARRANGEMENT

The hull exposure fire tests were conducted at the U. S. Coast Guard Fire and Safety Test Facility in Mobile, Alabama. The fires were located close along the aft starboard side of the tank ship MV RHODE ISLAND. The RHODE ISLAND is a T-1 tanker of steel construction, displacing 8500 gross tons and measuring 490 feet in length and 65 feet in beam. Figures 1 and 2 show a general profile of the starboard side and the location of the test area.

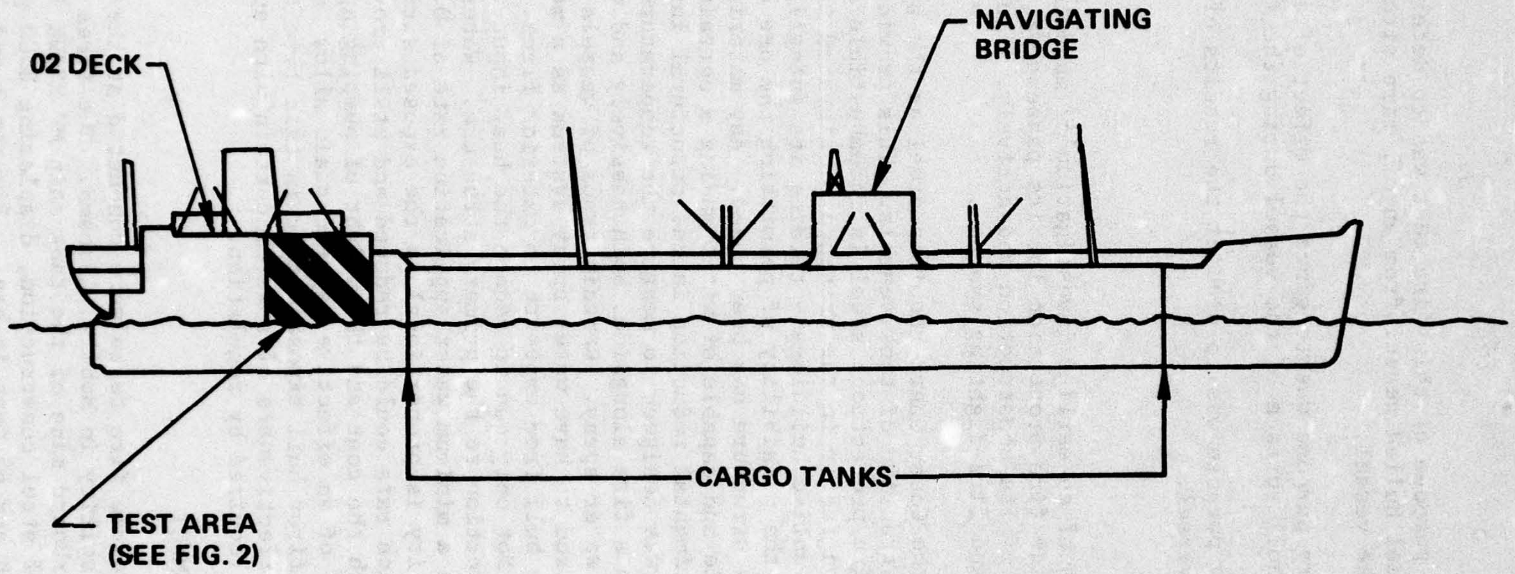


FIG. 1 GENERAL VESSEL OUTLINE

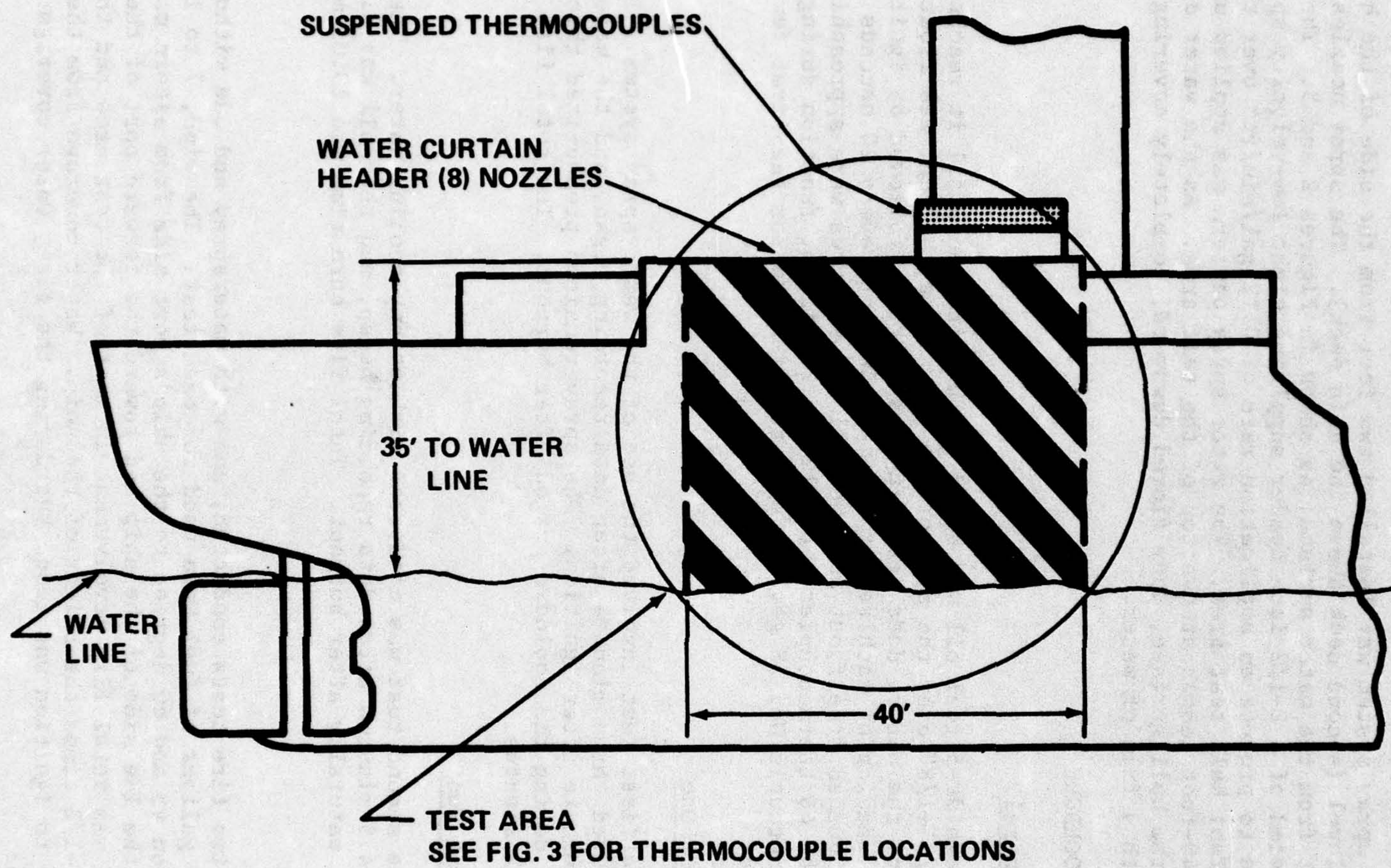


FIG. 2 FIRE TEST AREA

Fuel for the test fires was contained on the surface of the water alongside the ship by means of a floating steel oil-boom. This boom formed a crude semi-circle and had each of its ends secured to the hull so that the fuel could not leak out. The area thus formed measured approximately 2115 square feet.

A water spray system was installed two feet from the side of the hull at the 02 deck level (second deck above the main deck). The spray nozzles were about 35 feet from the water surface, as shwon in Figures 2 and 3. The spray system consisted of a 2-1/2 inch header supplying eight low-velocity spray nozzles sized to produce an application rate of 0.12 gal/min/ft^2 over the total 1400 square foot hull test area. The water spray pattern was applied uniformly over the 40-foot length at the top of the test area. As the water droplets impinged on the hull surface, they flowed downward, completely covering the test area with a film of water.

4.0 TEST PROCEDURE

4.1 General

Fresh JP-5 fuel oil was added to the fire pen until it reached a depth of 1-1/2"±1/8" over the area of the pen. Once the fuel was floated on the surface of the water, data recording was started, followed by ignition of the fire. A reading of each sensor was recorded once every 20 seconds until the fire had burned itself out and the hull temperatures were approaching ambient. Of the 46 thermocouples, Nos. 12 and 23 did not function during either test. (Thermocouple No. 28 gave intermittent results for the first few minutes of Test One.)

4.2 Test One

The first test involved the use of the water spray system. The fire was ignited three minutes after data recording began, and the water spray started one minute after ignition. The spray remained pressurized throughout the fire and during the cooldown period after burnout. The total fire burn time was 13.1 minutes.

4.3 Test Two

The second test was conducted without any cooling water. The fire was ignited 4.5 minutes after data recording began, and the hull was allowed to cool down naturally after burnout. Total fire burn time was 10.05 minutes.

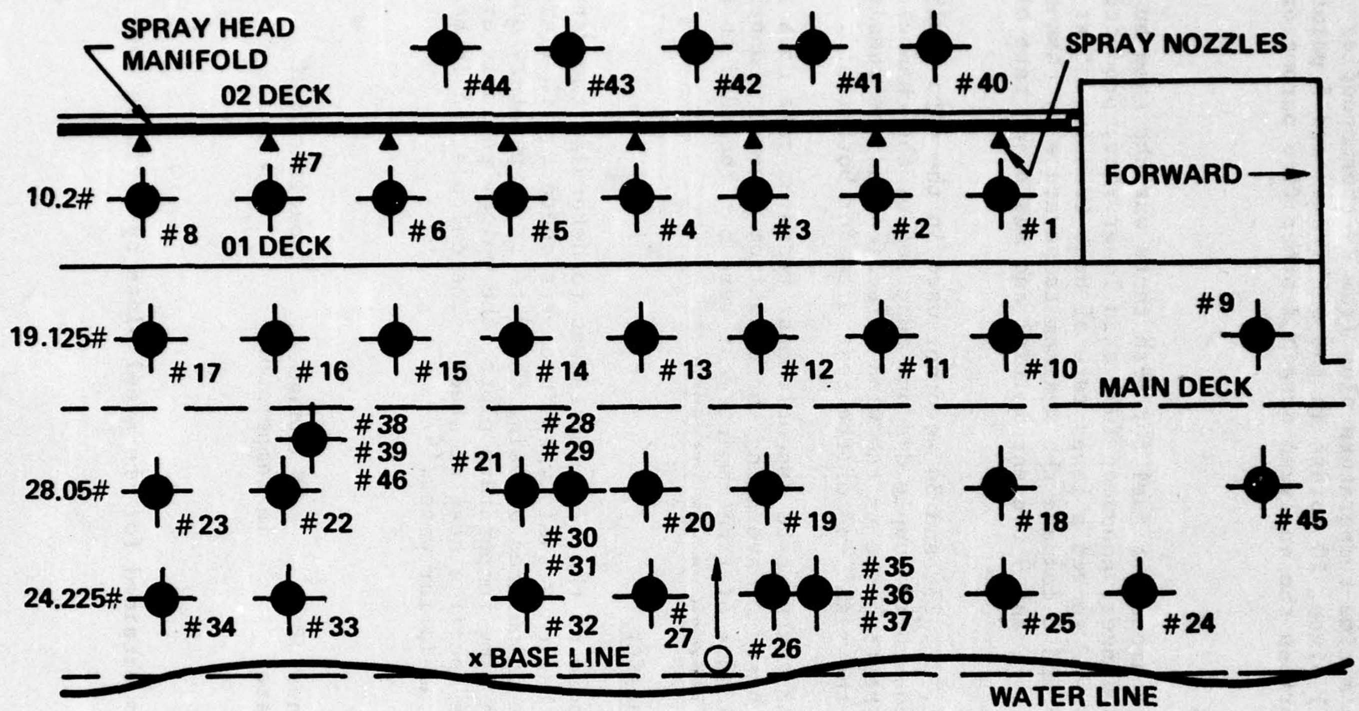
5.0 RESULTS

In the two fire tests conducted, one with waterspray and one without, two thousand gallons of fuel were used for each test. The wind, 7 to 11 MPH, varied between 45 and 60 degrees to the ship's port side from astern making the fire on the lee side of the ship and toward the forward part of the test area. There was total fire involvement in most of the test area and the flame height was 2-1/2 times the height of the ship. Water coverage from the spray system prior to ignition was good, but during the fire, water coverage was less then uniform.

#	x	y	#	x	y
1	31	21	24	3	16
2		26	25		22
3		30	26		30
4		34	27	3	34
5		38	28	11	37
6		42	29		
7		47	30		
8	31	52	31	11	37
9	20	11	32	3	38
10		21	33		47
11		26	34		52
12		30	35		29
13		34	36		
14		38	37	3	29
15		42	38	15	46
16		47	39	15	46
17	20	52	40	45	24
18	11	22	41		28
19		30	42		32
20		34	43		36
21		38	44	45	40
22		47	45	11	11
23	11	52	46	15	46

THERMOCOUPLE
COORDINATES

# = THERMO#
x = DISTANCE FM
x BASE LINE
y = DISTANCE FM
y BASE LINE



x & y DIMENSIONS ARE IN FEET

STBD. SIDE, OUT BOARD

FIG. 3 THERMOCOUPLE GRID IN TEST AREA (NOT TO SCALE)

The recorded data was a time-temperature plot from 29 thermocouples imbedded in the ship's hull plate, 10 points in the web frames and 5 points in the air. Table 1 summarizes the analyzed data and other fire parameters.

6.0 DISCUSSION OF RESULTS

Table 1 shows that the amount of fuel for both tests was the same but that the first fire burned longer, consumed fuel at a lower rate, produced lower average temperatures and showed a lower rate of heat transfer after data reduction. Figures 5 and 6 compare the maximum temperature by channel for both tests. Figures 7, 8, and 9 present initial and sustained rate of heat transfer.

Channels 7, 8, 12, 17, 23, 27, and 34 were not used in the data analysis for one of two reasons: the thermocouple did not function or the channel showed little evidence of heating due to the wind direction and the result that the fire burned near the right end of the test area for both tests.

Since a purpose of the tests is to assemble heat transfer rate data and the method of analysis is based on averaging the data from several channels, channels without clear evidence of fire impingement were not included to avoid adverse distortion of the average values presented.

6.1 Rate of Heat Transfer

The primary purpose of the test series was to determine the rate of heat transfer to steel hulled vessels when in direct contact with flames. The incident heat flux was calculated by using the rate of temperature change of the hull plate from the time temperature plots, the weight per unit area of the hull plate, and the specific heat of steel. The calculation is based on the approximate relationship for water:[4]

$$Q(\text{BTU}) = \begin{matrix}\text{pounds of}\\ \text{water}\end{matrix} \quad \begin{matrix}{}^\circ\text{F change}\\ \text{in temperature}\end{matrix} \quad \begin{matrix}\text{specific heat}\\ \text{of water}\end{matrix}$$

A rate of heat transfer is obtained for the steel plate by dividing both sides by time and area:

TABLE 1

RESULTS OF TESTS

	TEST 1	TEST 2
AMOUNT OF FUEL	2000 gallons	2000 gallons
BURN TIME	13.1 minutes	10.05 minutes
BURN RATE (depth of fuel)	0.12 in/min 160 gallons/min	0.15 in/min 200 gallons/n
DATA RECORDING TO IGNITION, TIME LAG	3.0 minutes	4.5 minutes
IGNITION TO WATER SPRAY, TIME LAG	1.0	(no water spray
WIND SPEED	7 MPH 3 m/s	11 MPH 5 m/s
Average of Maximum Temperatures	305°C 581°F	325°C 617°F
Maximum Temperature of any channel	637°F 1179°F	632°C 1170°F
Initial rate of heat transfer	79%	100%
Sustained rate of heat transfer	70%	100%
Maximum Air Temperature	336°C/637°F	850°C/1563°F
Average of 5 Maximum Air Temperatures	171°C/340°F	590°C/1094°F
AVERAGE MAXIMUM TEMPERATURE		
Row 1	82°C/180°F	329°C/624°F
Row 2	219°C/426°F	220°C/428°F
Row 3	426°C/864°F	344°C/651°F
Row 4	505°C/941°F	446°C/835°F

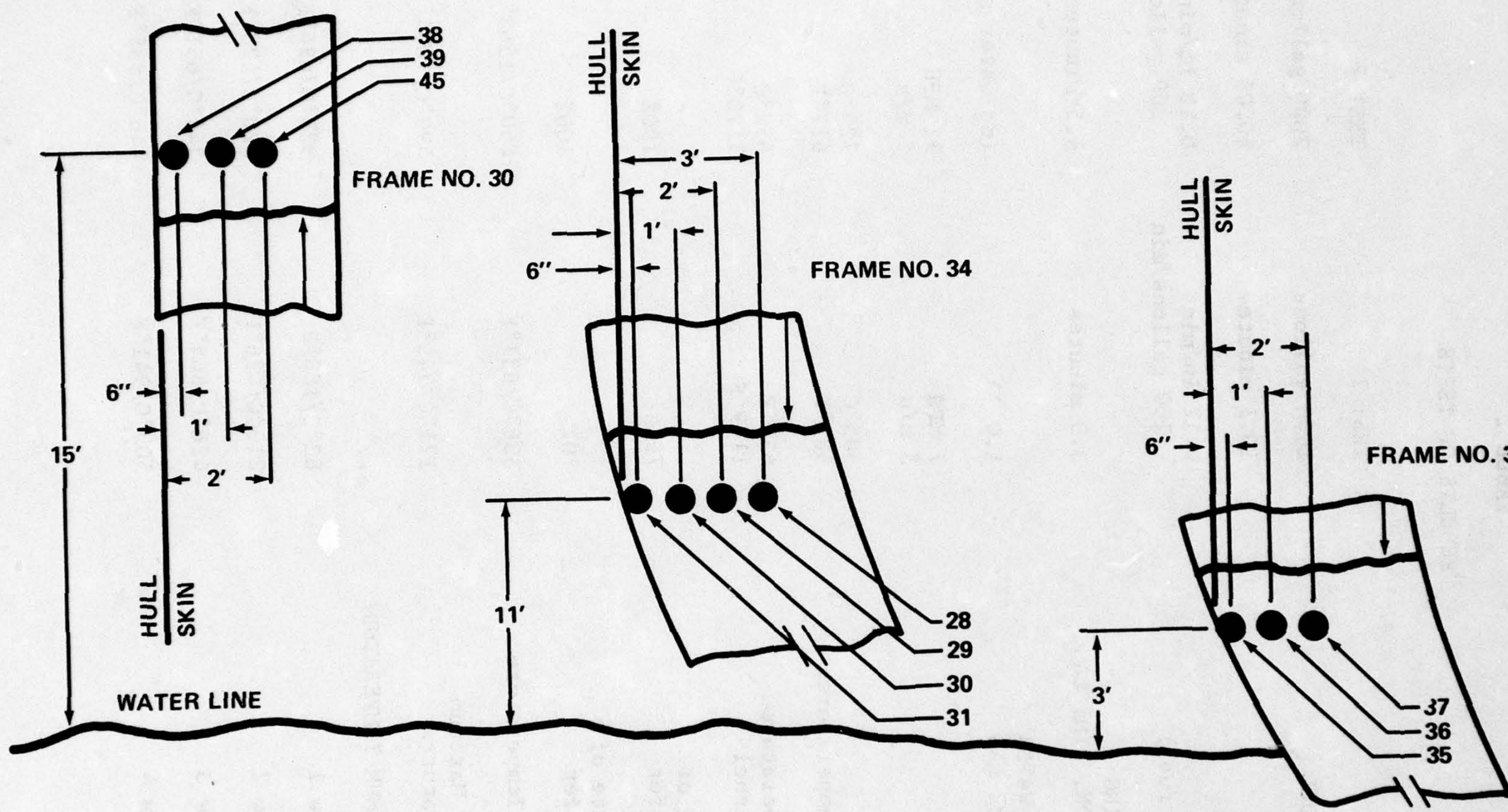
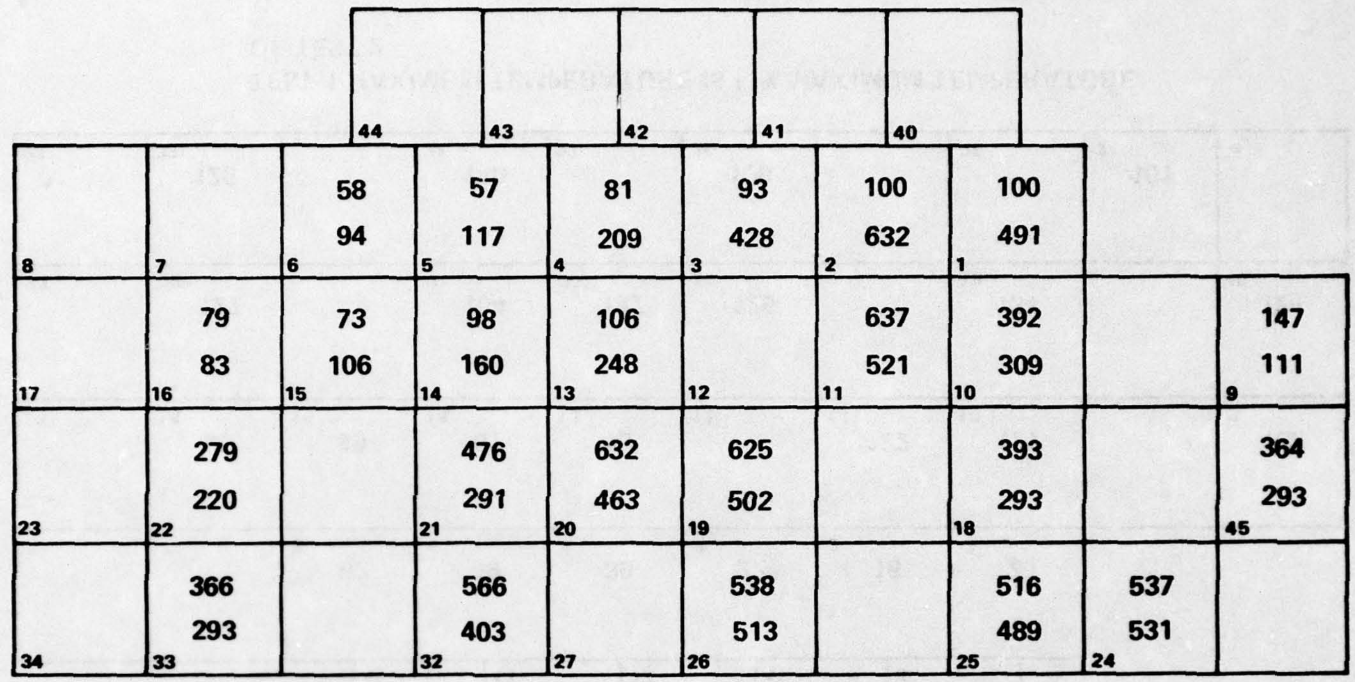
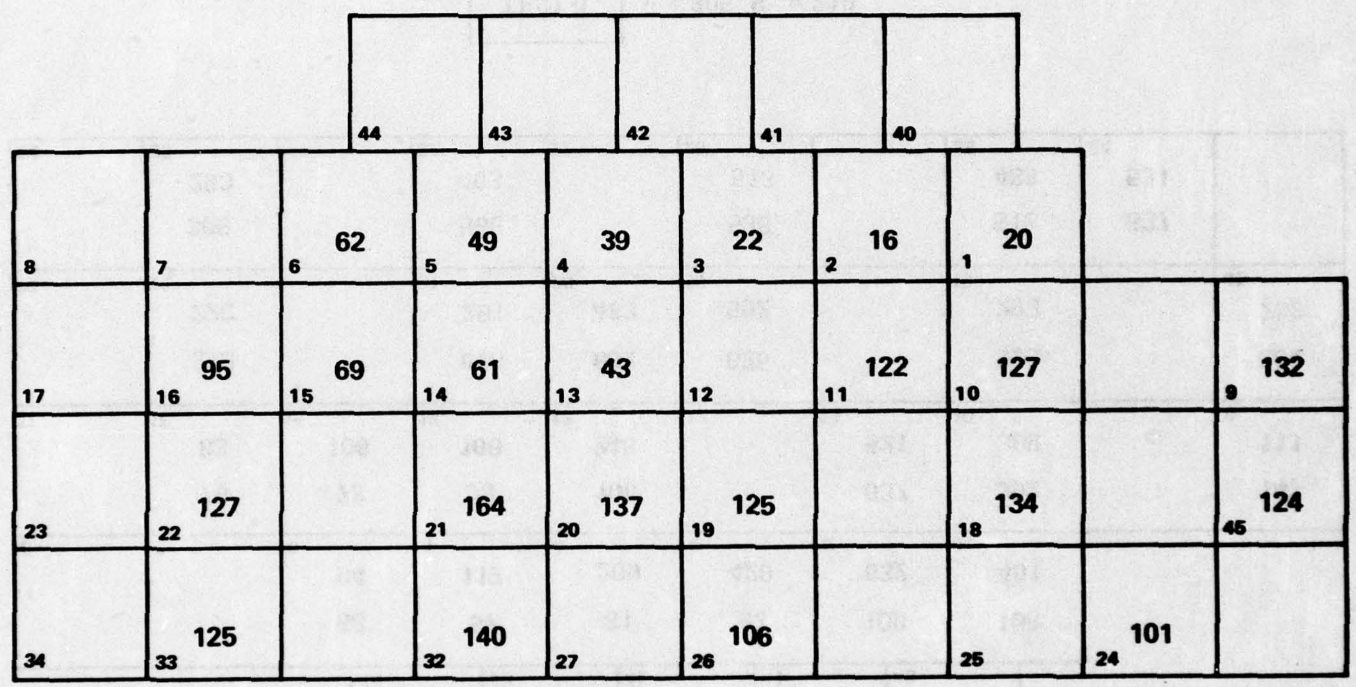


FIG. 4 THERMOCOUPLE LOCATIONS IN WEB FRAMES (NOT TO SCALE)



TEST 1	$\bar{x} = 305$ $S_x = 219$
TEST 2	$\bar{x} = 325$ $S_x = 167$

FIG. 5 MAXIMUM TEMPERATURE ATTAINED °C



TEST 1 MAXIMUM TEMPERATURE IS ( )% MAXIMUM TEMPERATURE OF TEST 2

FIG. 6 MAXIMUM TEMPERATURE COMPARISON BY PERCENTAGE

		44	43	42	41	40			
8	7	6: 1400 1000	5: 800 1800	4: 8200 14000	3: 11500 33700	2: 7700 32800	1: 11700 36700		
17	16: 1100 1400	15: 3200 3700	14: 3600 9300	13: 4200 14200	12	11: 25600 42500	10: 11300 16300		9: 2800 2200
23	22: 20400 10900		21: 34500 23000	20: 47900 41200	19: 33300 41200		18: 15000 18200		45: 72000 95300
34	33: 13500 17200		32: 26800 27600	27	26: 22000 44800		25: 23900 30800	24: 25100 32900	

TEST 1 $\overline{x} = 17800$ $S_x = 17000$
TEST 2 $\overline{x} = 24700$ $S_x = 20900$

TEST 1
TEST 2

IN $\frac{BTU}{HR\ Ft^2}$

$$\frac{\overline{x}_1}{\overline{x}_2} = 72\%$$

FIG. 7 INITIAL HEAT TRANSFER RATE

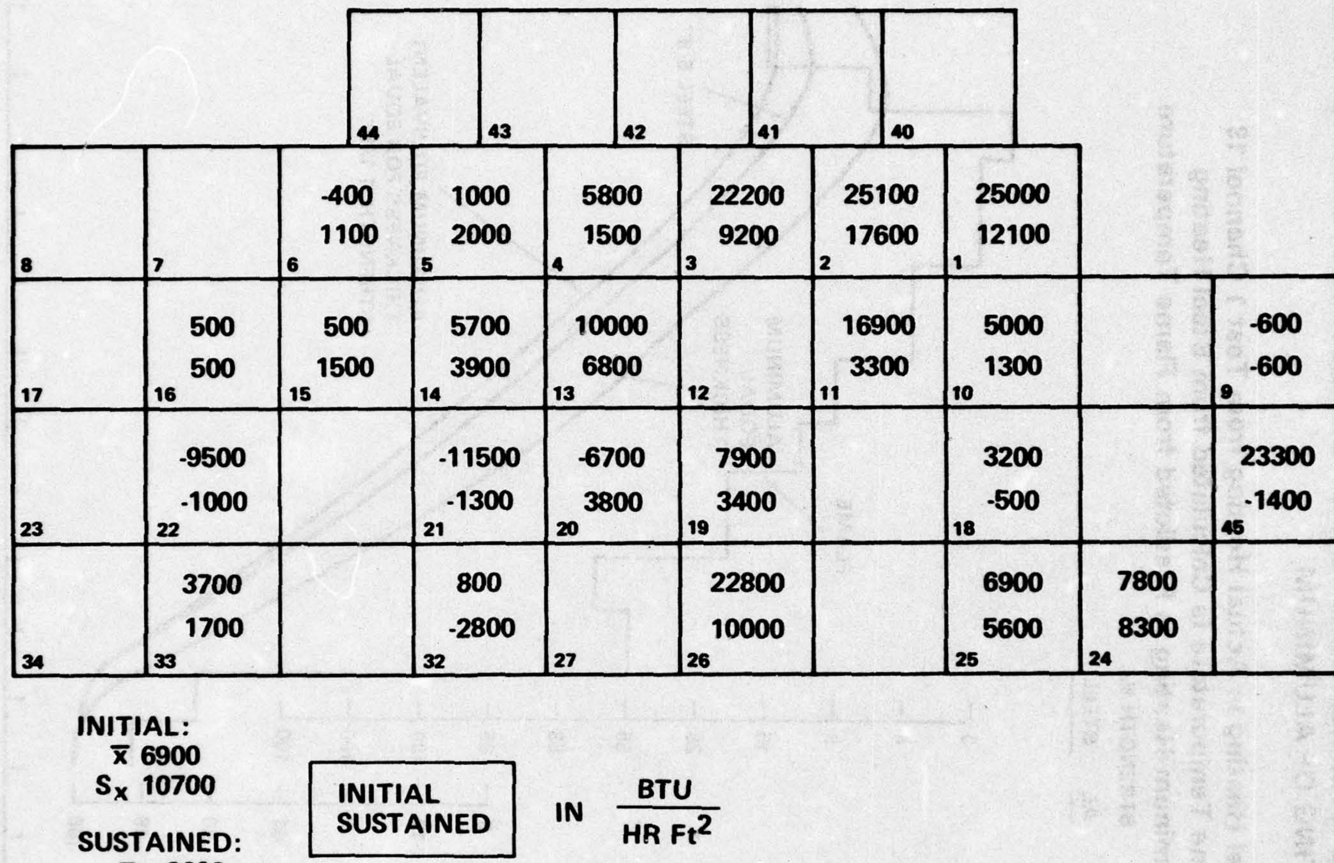
		44	43	42	41	40			
8	7	-100 1000 6	-200 1800 5	2900 4400 4	4900 14100 3	1900 19500 2	900 13000 1		
17	1100 1400 16	900 2400 15	1000 4900 14	1800 8600 13	12	15800 19100 11	9100 10400 10		2800 2200 9
23	9000 8000 22		14400 13100 21	20000 23800 20	20400 23800 19		12300 11800 18		14100 12700 45
34	12600 14300 33		21700 18900 32	27	17000 27000 26		17200 22800 25	17200 25500 24	

TEST 1 $\bar{x} = 9100$ $S_x = 7700$
TEST 2 $\bar{x} = 12700$ $S_x = 8400$

TEST 1
TEST 2

IN $\frac{BTU}{HR\ FT^2}$ $\frac{\bar{x}_1}{\bar{x}_2} = 72\%$

FIG. 8 SUSTAINED HEAT TRANSFER RATE



INITIAL:
$\overline{x}$ 6900
S_x 10700

SUSTAINED:
$\overline{x} = 3600$
$S_x = 4900$

FIG. 9 REDUCTION IN HEAT INPUT RATE FOR TEST 2 MINUS RATE FOR TEST 1

FIGURE 10. STEEL HULL HEATING, FLAME TEMPERATURE AND PREDICTED HEATING OF ALUMINUM

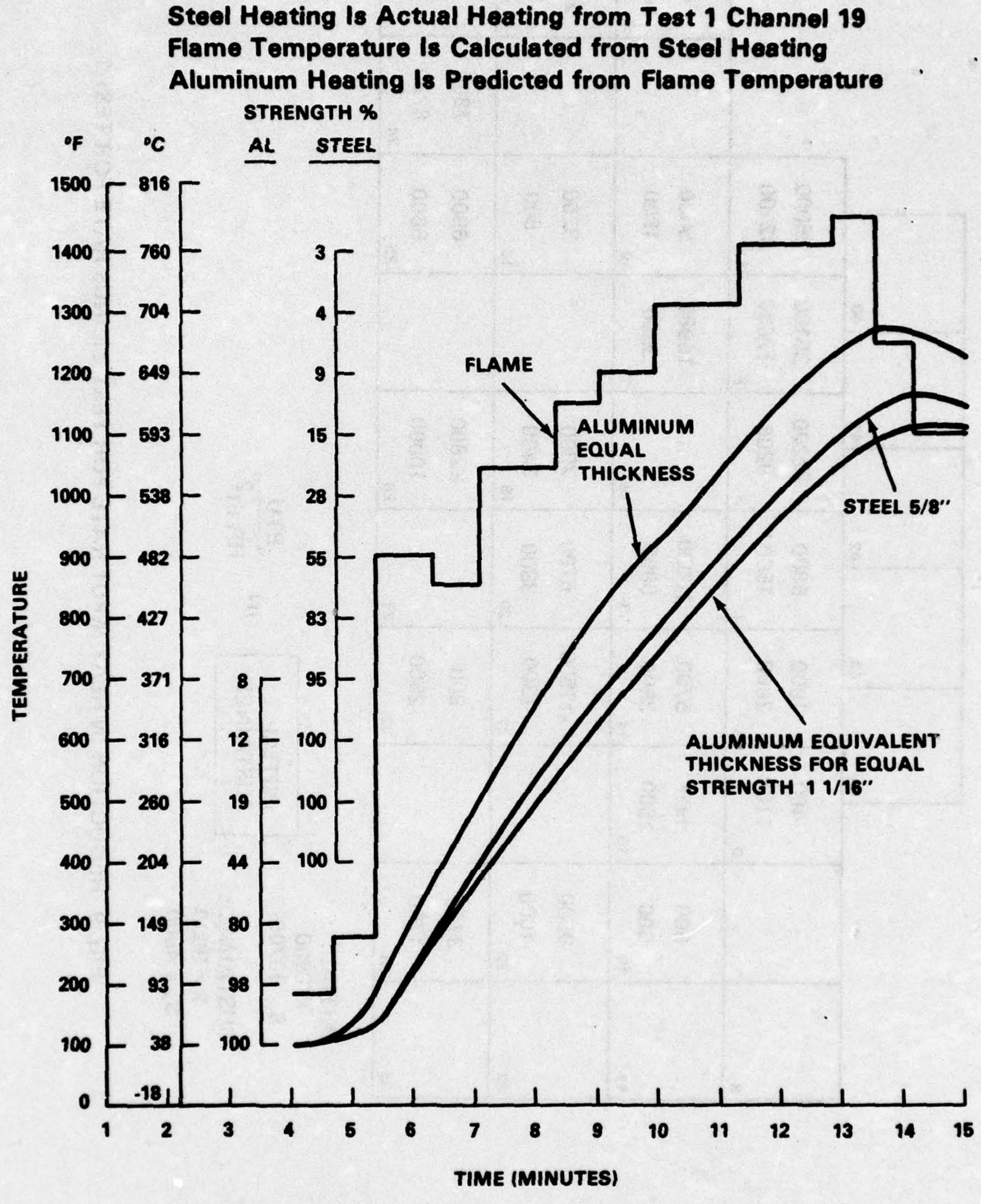


FIGURE 11. STEEL HULL HEATING, FLAME TEMPERATURE AND PREDICTED HEATING OF ALUMINUM

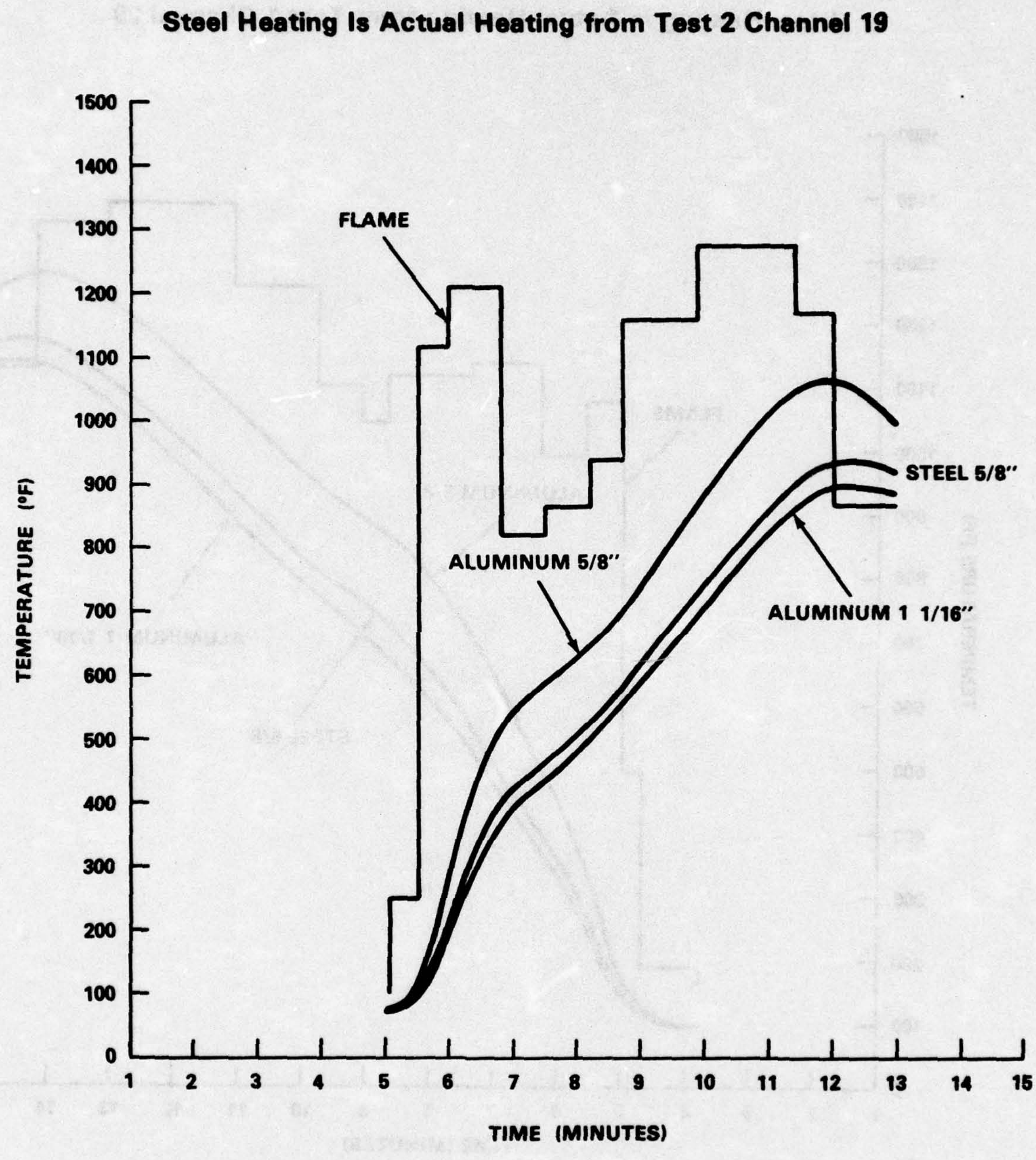


FIGURE 12. STEEL HULL HEATING, FLAME TEMPERATURE, AND PREDICTED HEATING OF ALUMINUM

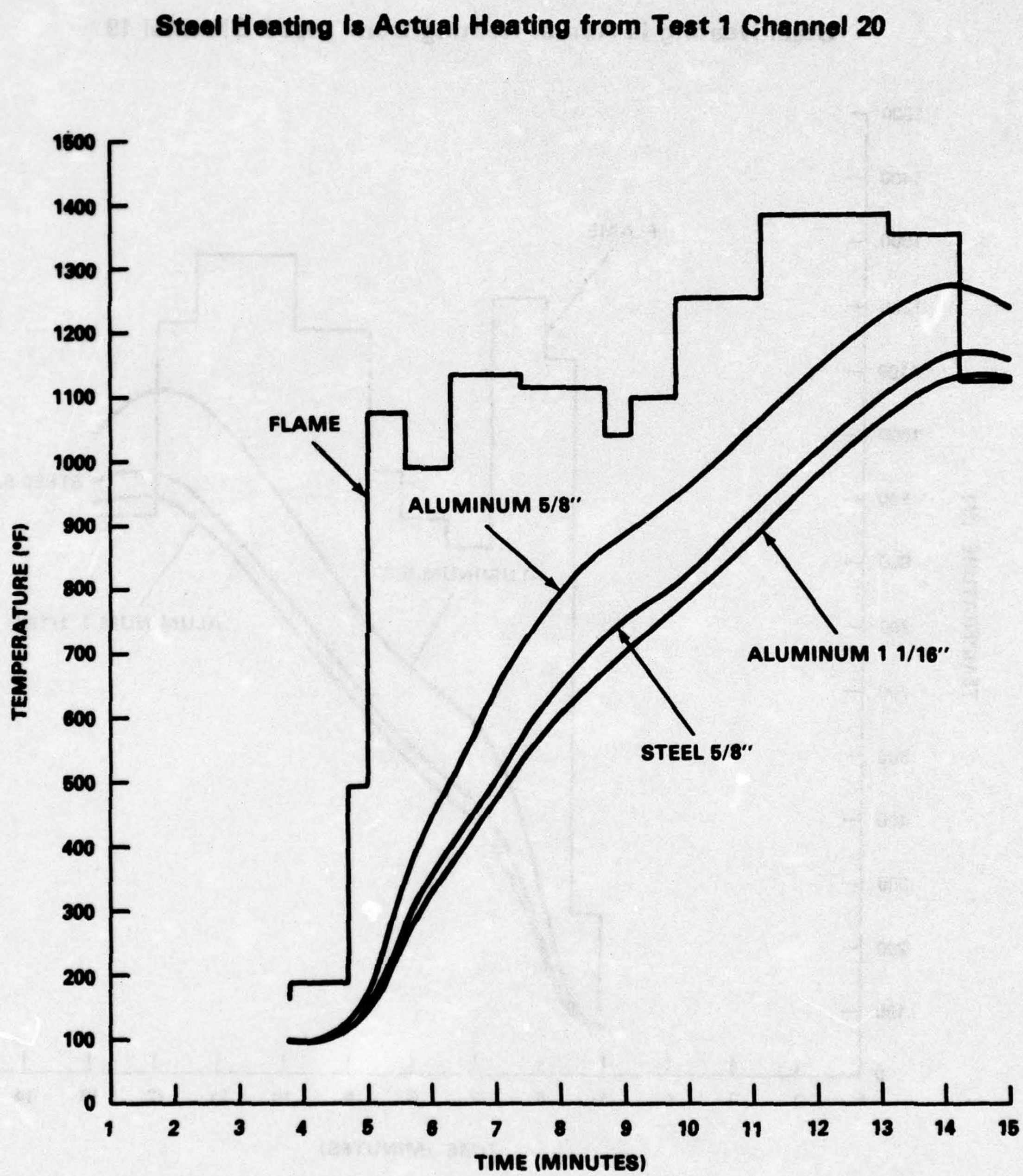
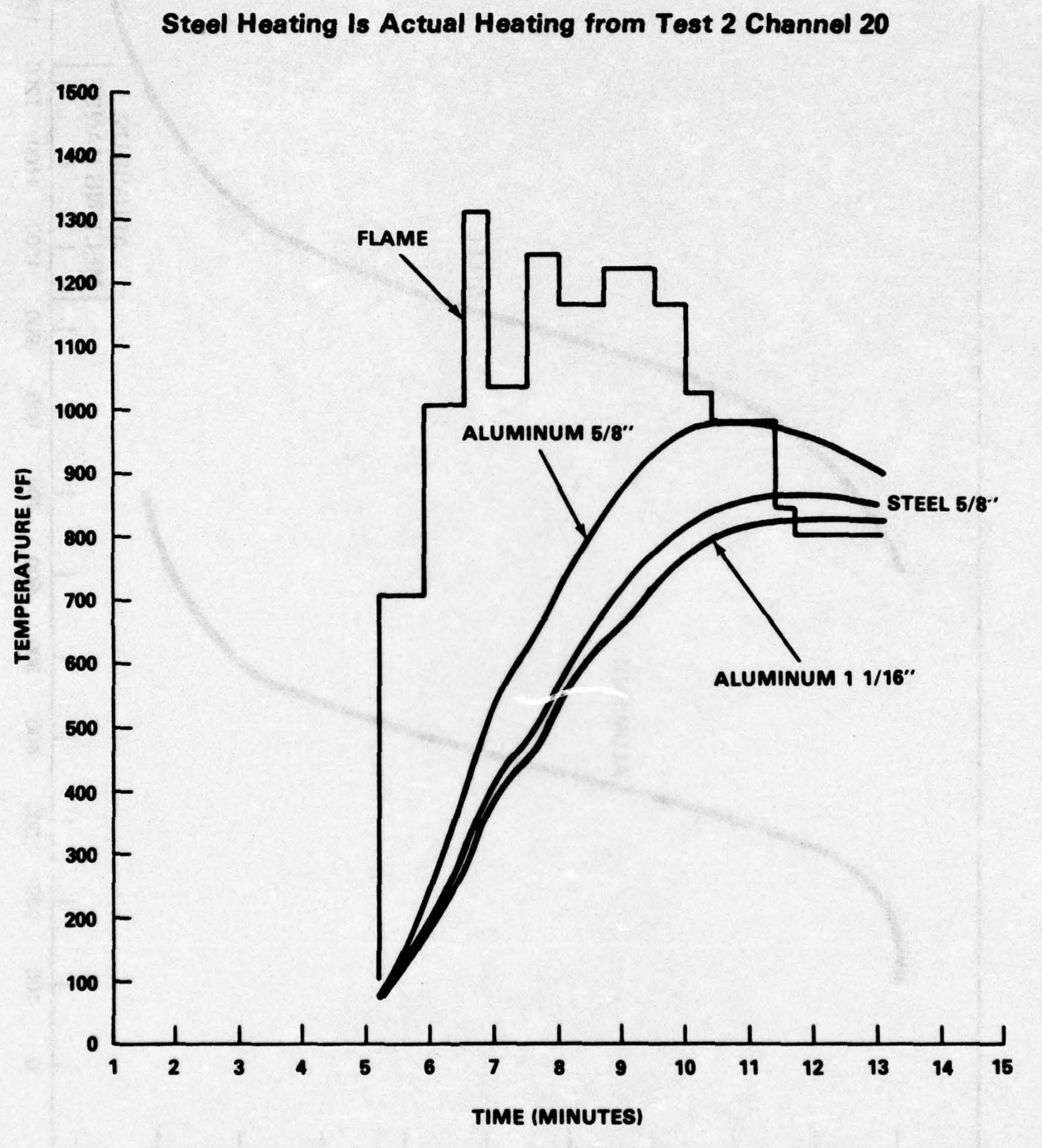


FIGURE 13. STEEL HULL HEATING, FLAME TEMPERATURE, AND PREDICTED HEATING OF ALUMINUM



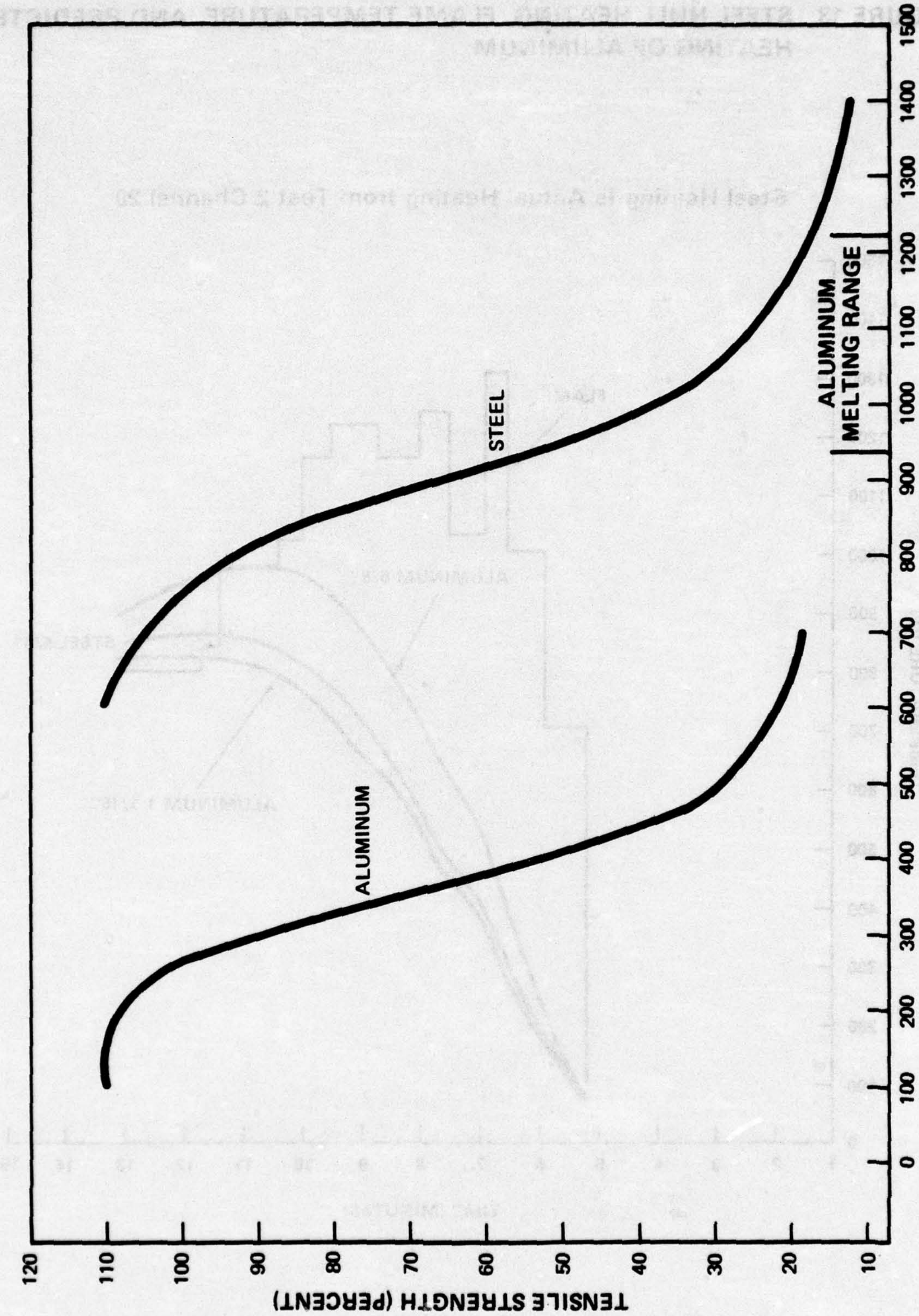


FIG. 14 TYPICAL STRENGTH vs TEMPERATURE

FIGURE 15. PLATE HEATING ACTUAL AND PREDICTED

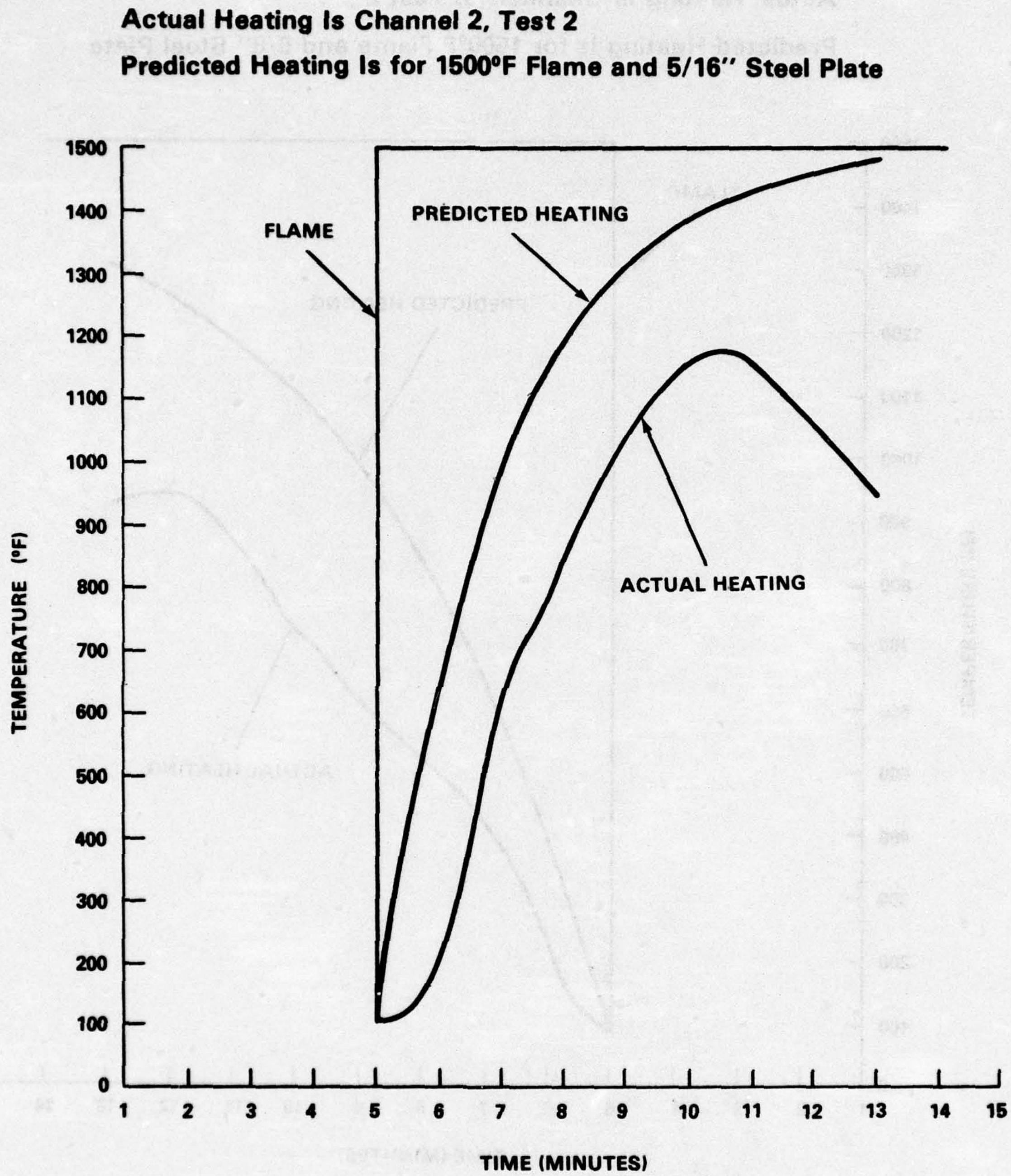


FIGURE 16. PLATE HEATING - ACTUAL AND PREDICTED

Actual Heating Is Channel 19, Test 2

Predicted Heating Is for 1500°F Flame and 5/8" Steel Plate

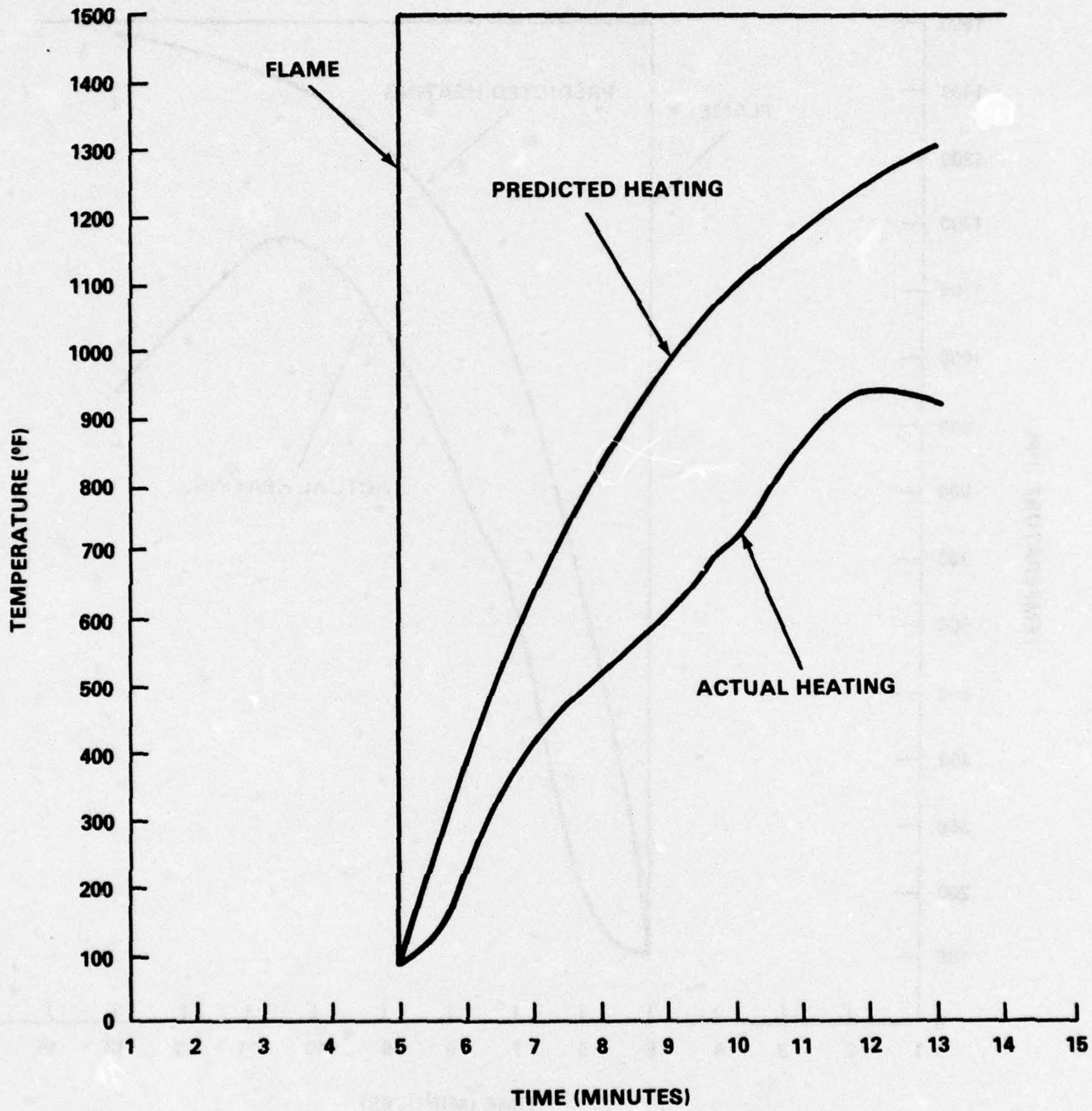
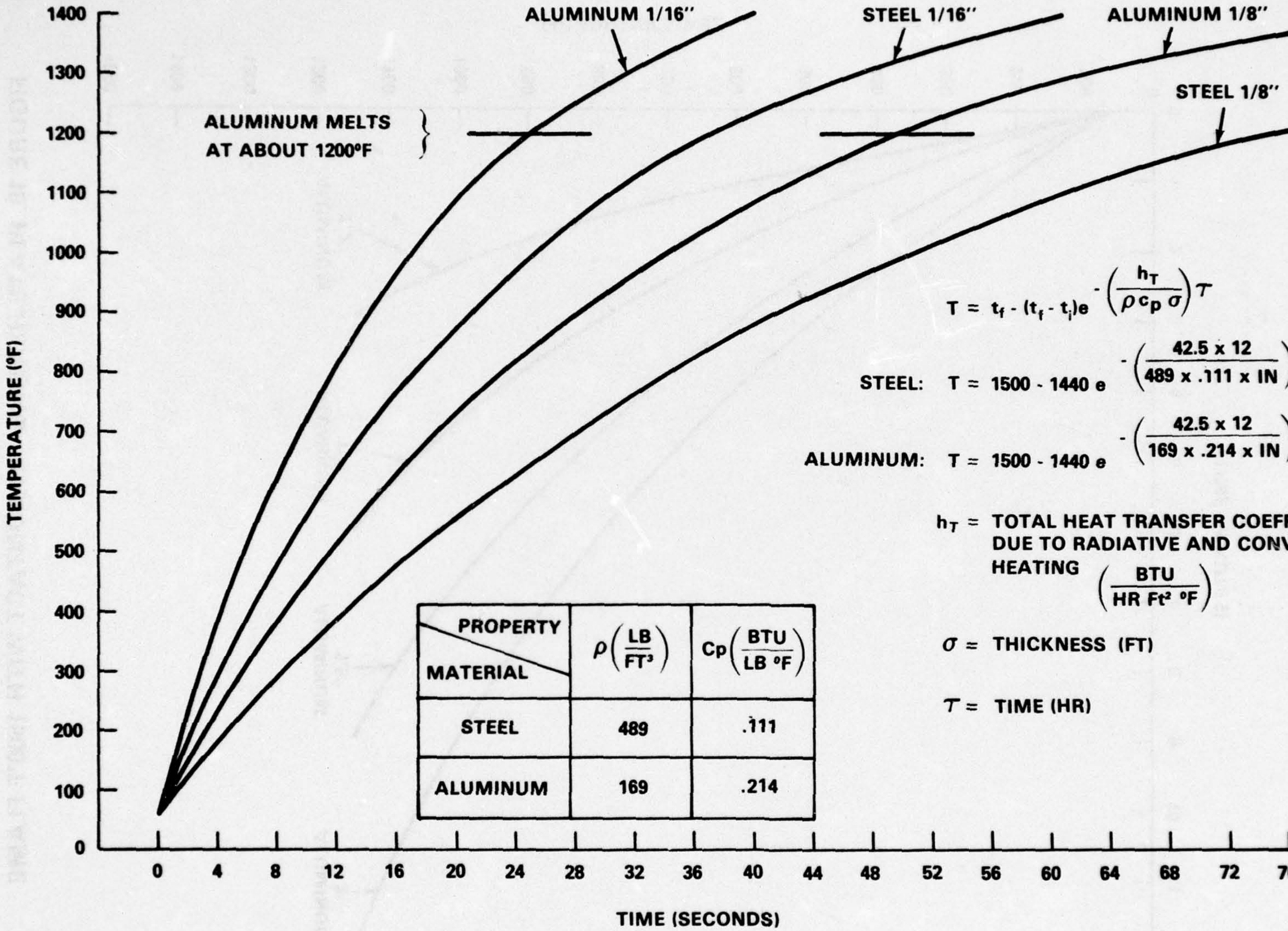


FIGURE 17. PLATE HEATING - IN CONTACT WITH 1500°F FLAME



PROPERTY / MATERIAL	$\rho\left(\frac{LB}{FT^3}\right)$	$Cp\left(\frac{BTU}{LB\ °F}\right)$
STEEL	489	.111
ALUMINUM	169	.214

$$T = t_f - (t_f - t_i)e^{-\left(\frac{h_T}{\rho c_p \sigma}\right)\tau}$$

STEEL: $$T = 1500 - 1440\, e^{-\left(\frac{42.5 \times 12}{489 \times .111 \times IN}\right)\frac{\tau}{3600}}$$

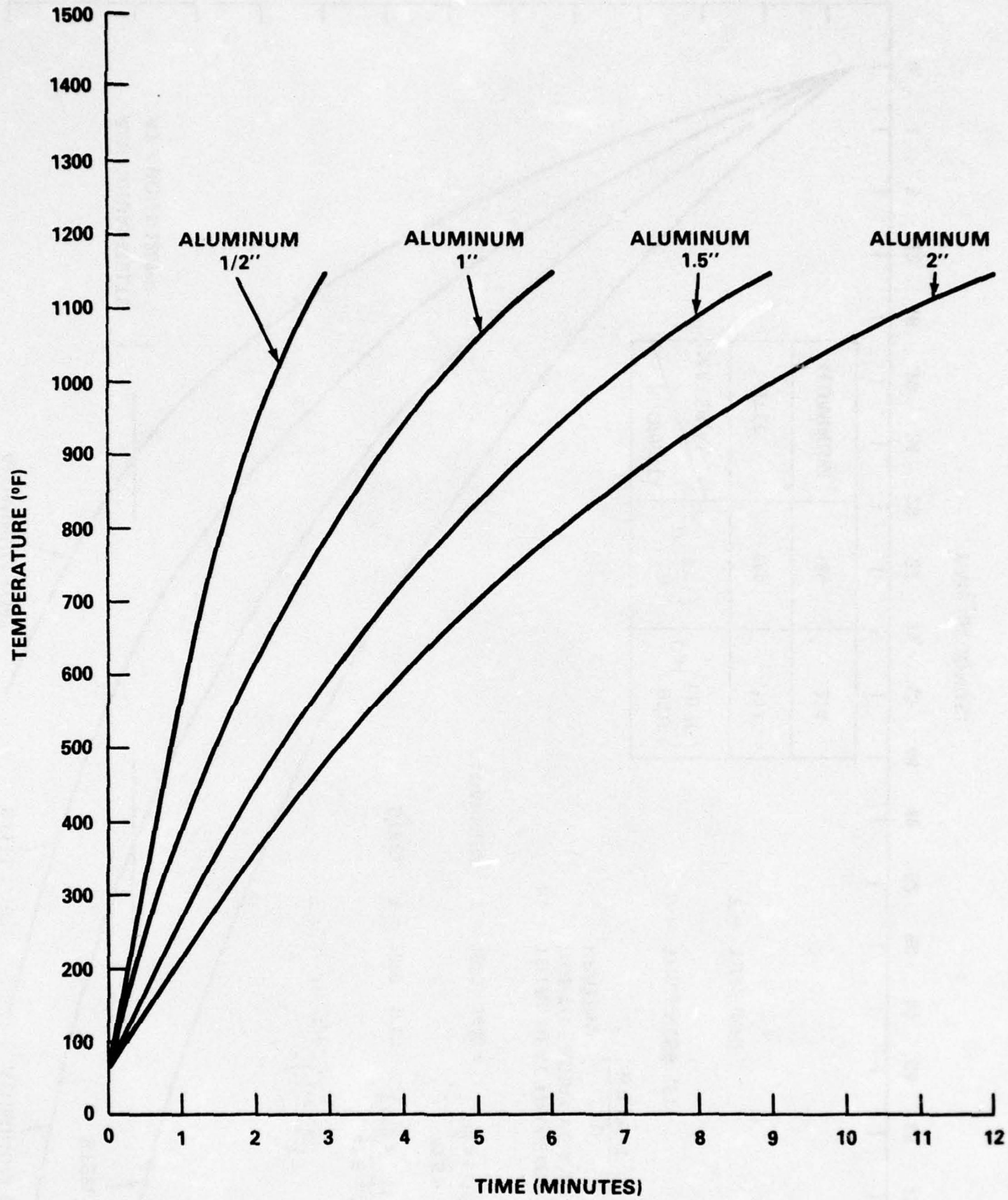
ALUMINUM: $$T = 1500 - 1440\, e^{-\left(\frac{42.5 \times 12}{169 \times .214 \times IN}\right)\frac{\tau}{3600}}$$

h_T = TOTAL HEAT TRANSFER COEFFICIENT DUE TO RADIATIVE AND CONVECTIVE HEATING $\left(\frac{BTU}{HR\ Ft^2\ °F}\right)$

σ = THICKNESS (FT)

τ = TIME (HR)

FIGURE 18. PLATE HEATING - IN CONTACT WITH 1500°F FLAME



$$q\ (\text{BTU/hr ft}^2) = \frac{\text{pounds of steel}}{\text{ft}^2} \quad \frac{\Delta T\ \text{change in temperature}}{\text{hr}} \quad \text{specific heat of steel}$$

This is the same as:

q = plate weight x slope of graph x factor

The factor is composed of the specific heat of steel times appropriate conversion factors to adjust the slope of the graph from °C/minutes to °F/hr. Finally:

$$q = W \times \frac{\Delta T}{\Delta t} \times 13.7 \quad \text{for steel}$$

where W is the plate weight (lbs/ft^2)

T is the temperature (°C)

t is the time (min)

q is the heat transfer rate (BTU/hr. ft^2)

This relationship can be used on Figures 19 through 26, which are selected channels from both tests of the fires.

The plate weights were obtained by drilling a hole through the hull for each of the four rows of thermocouples and gaging the plate thickness. Since one inch thick plate weights 40.8 lbs/ ft^2, the plate weight for any measured thickness is easily determined.

The measured thicknesses are given in Table 2.

TABLE 2

HULL PLATE THICKNESS

LOCATION	THICKNESS	lbs./ft^2
Between 02 deck and 01 deck	5/16"	12.7
Between 01 deck and main deck	7/16"	17.8
Below main deck	5/8"	25.5
Above water line	5/8"	25.5

This method of calculating the incident heat flux assumes that the temperature gradient within the plate and heat losses to the inside of the ship are negligible. These assumptions are addressed later. The time temperature plots were digitized by placing the plots on a coordinate sensing table and tracing the plot with a sensor. A computer connected to both the table and the sensor digitized and printed coordinate values representative of the plots. From these coordinate values the slope of the plot was calculated and from the previous equation the rate of heat transfer was calculated. Figures 7, 8, and 9 present the results.

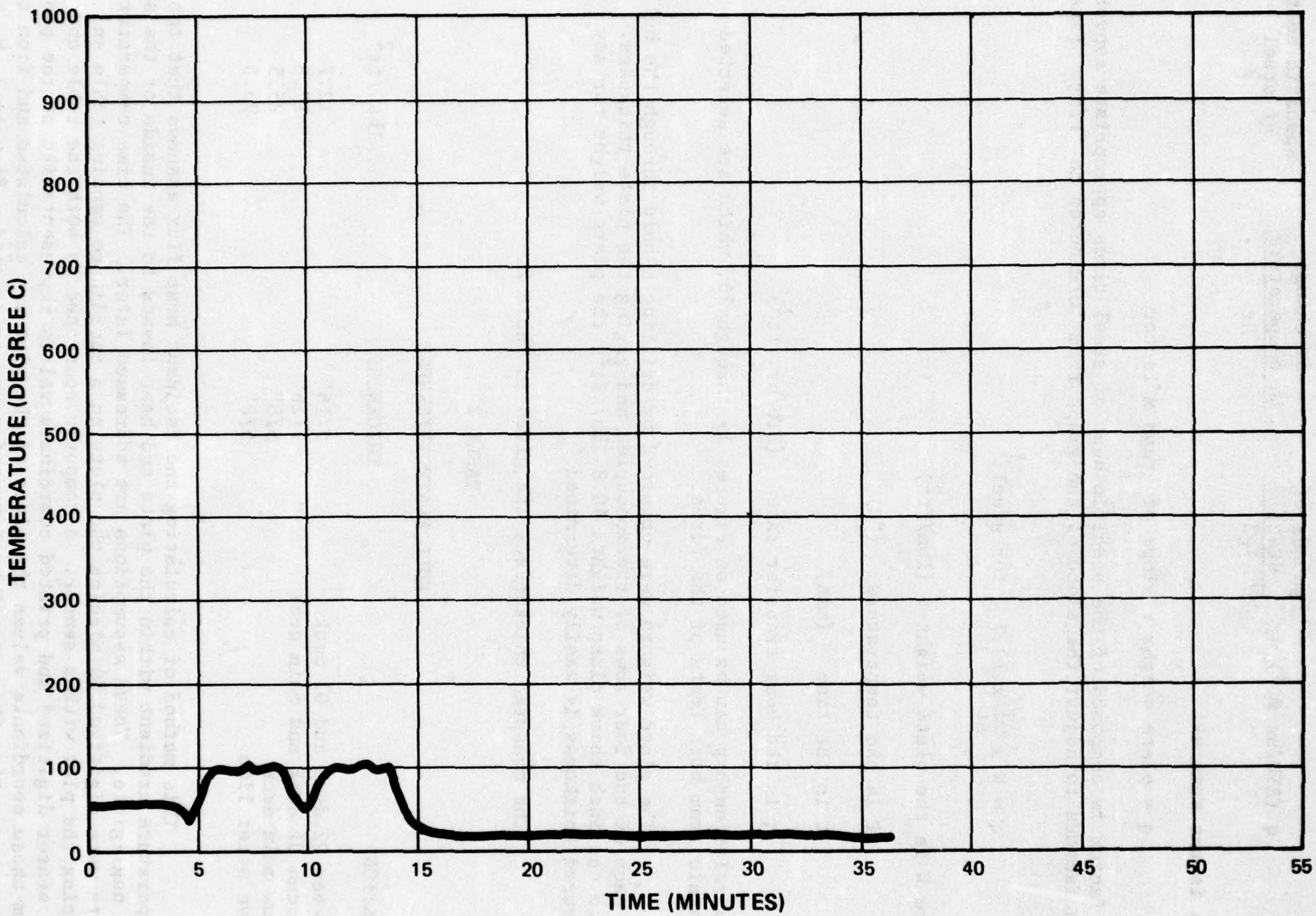


FIG. 19 HULL EXPOSURE FIRE TEST 1
CHANNEL 2

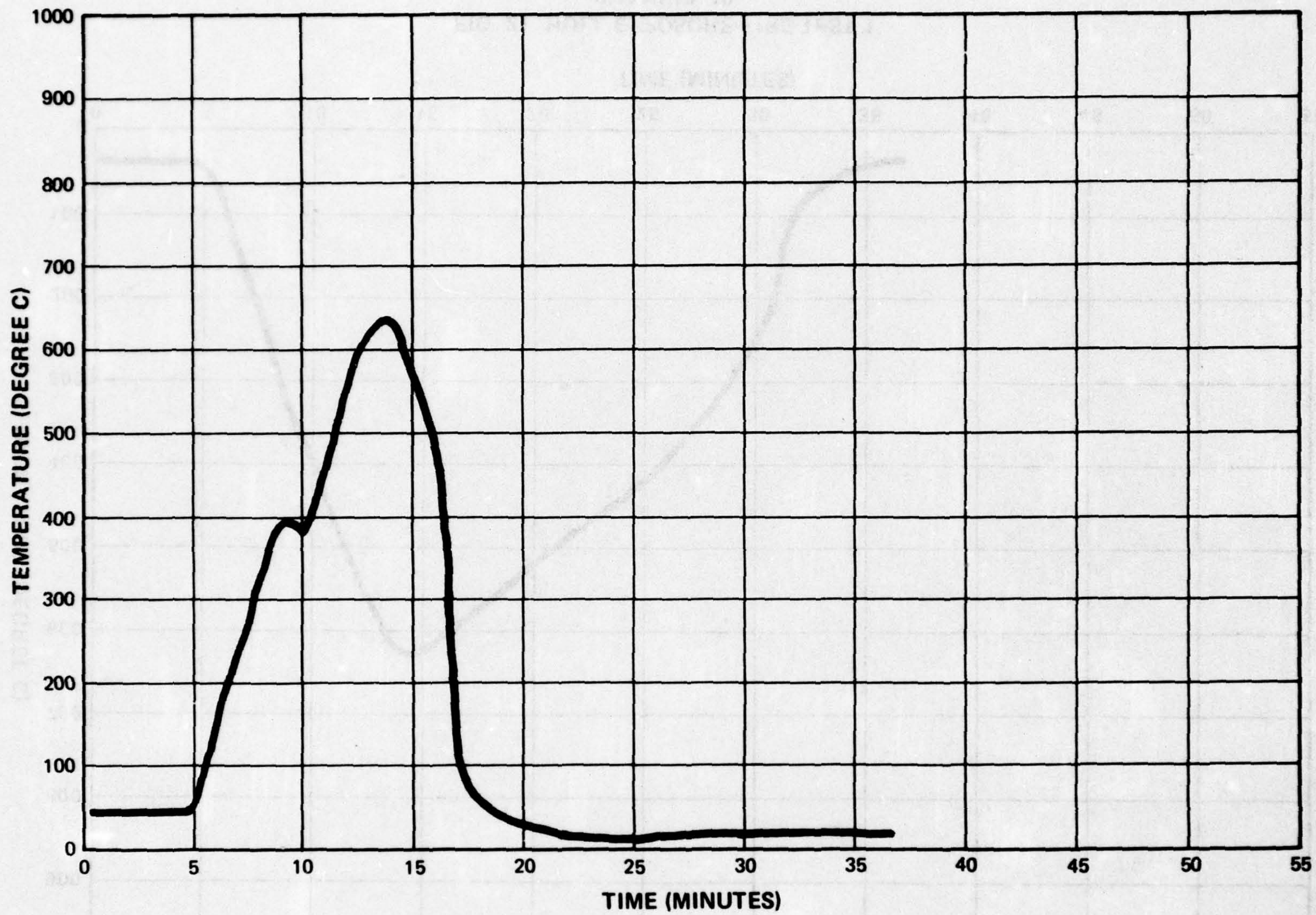


FIG. 20 HULL EXPOSURE FIRE TEST 1
CHANNEL 11

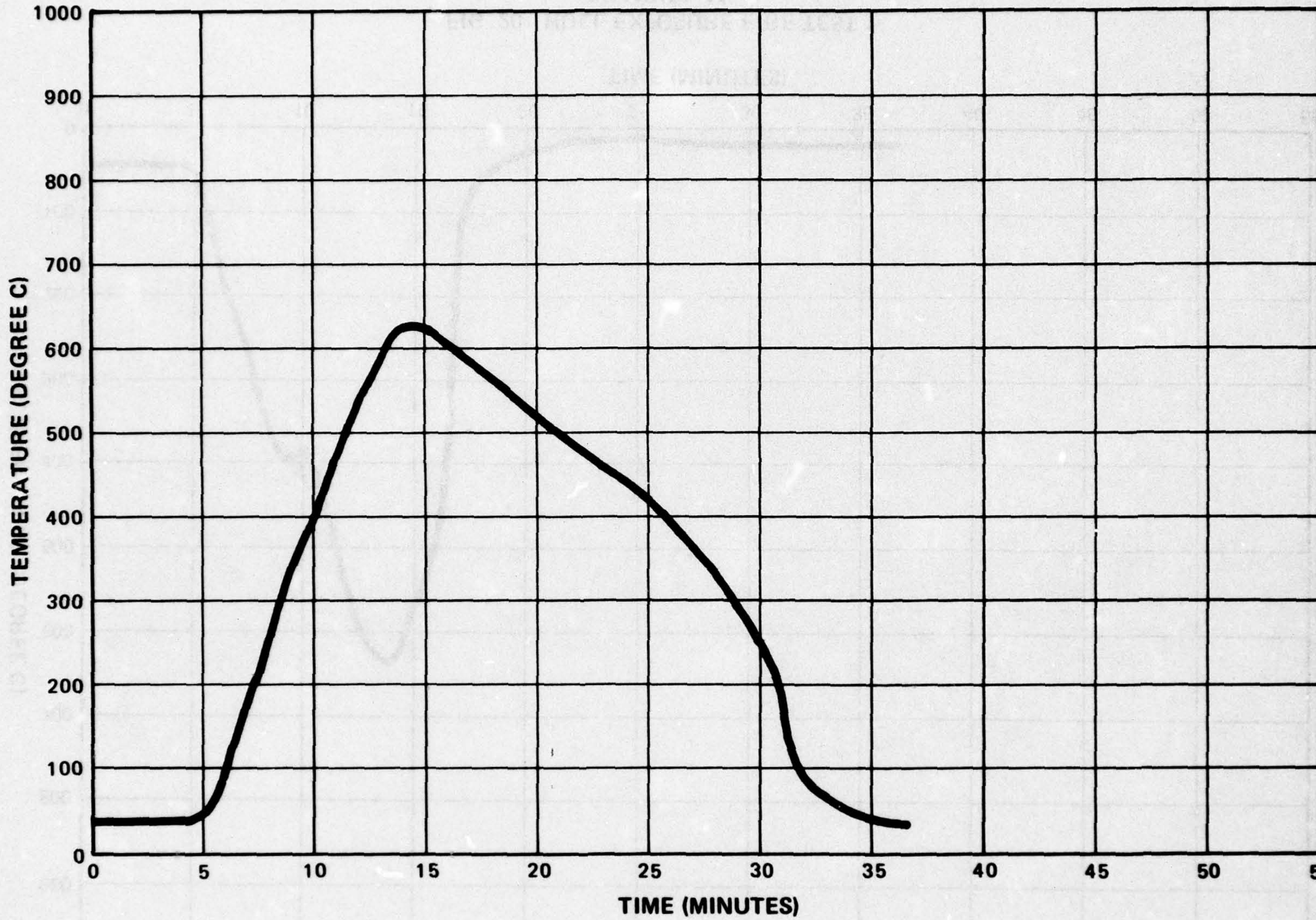


FIG. 21 HULL EXPOSURE FIRE TEST 1
CHANNEL 19

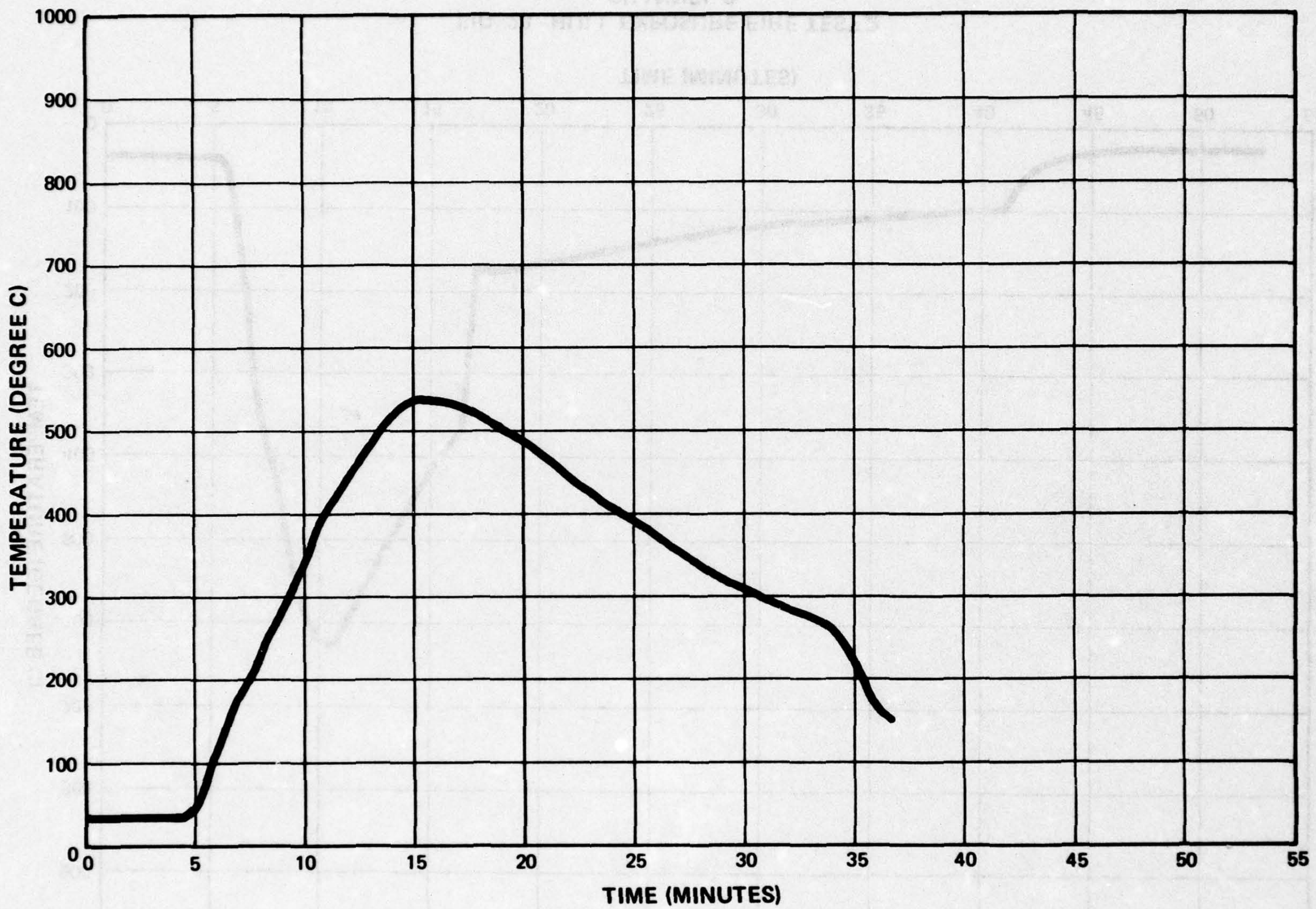


FIG. 22 HULL EXPOSURE FIRE TEST 1
CHANNEL 26

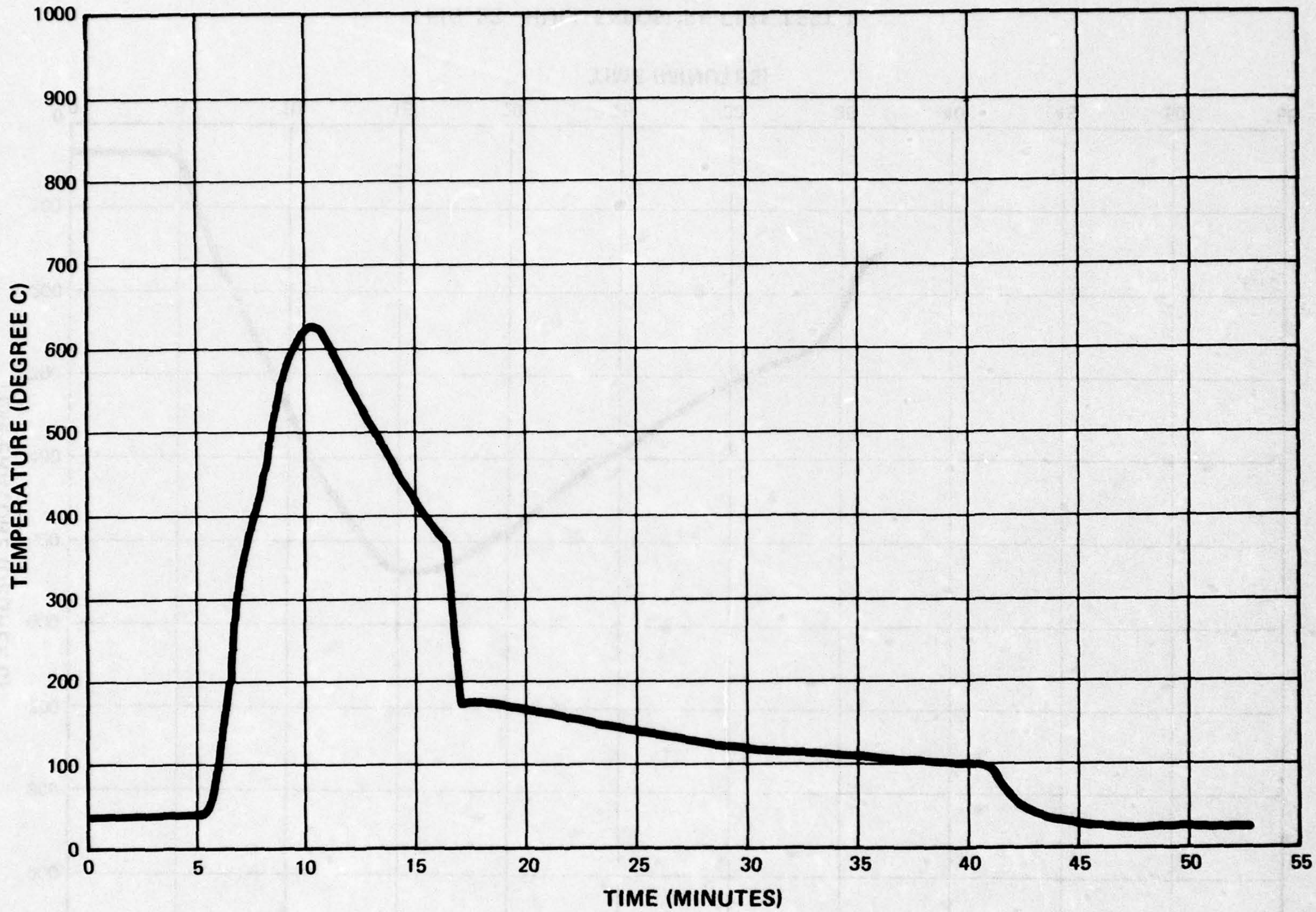


FIG. 23 HULL EXPOSURE FIRE TEST 2
CHANNEL 2

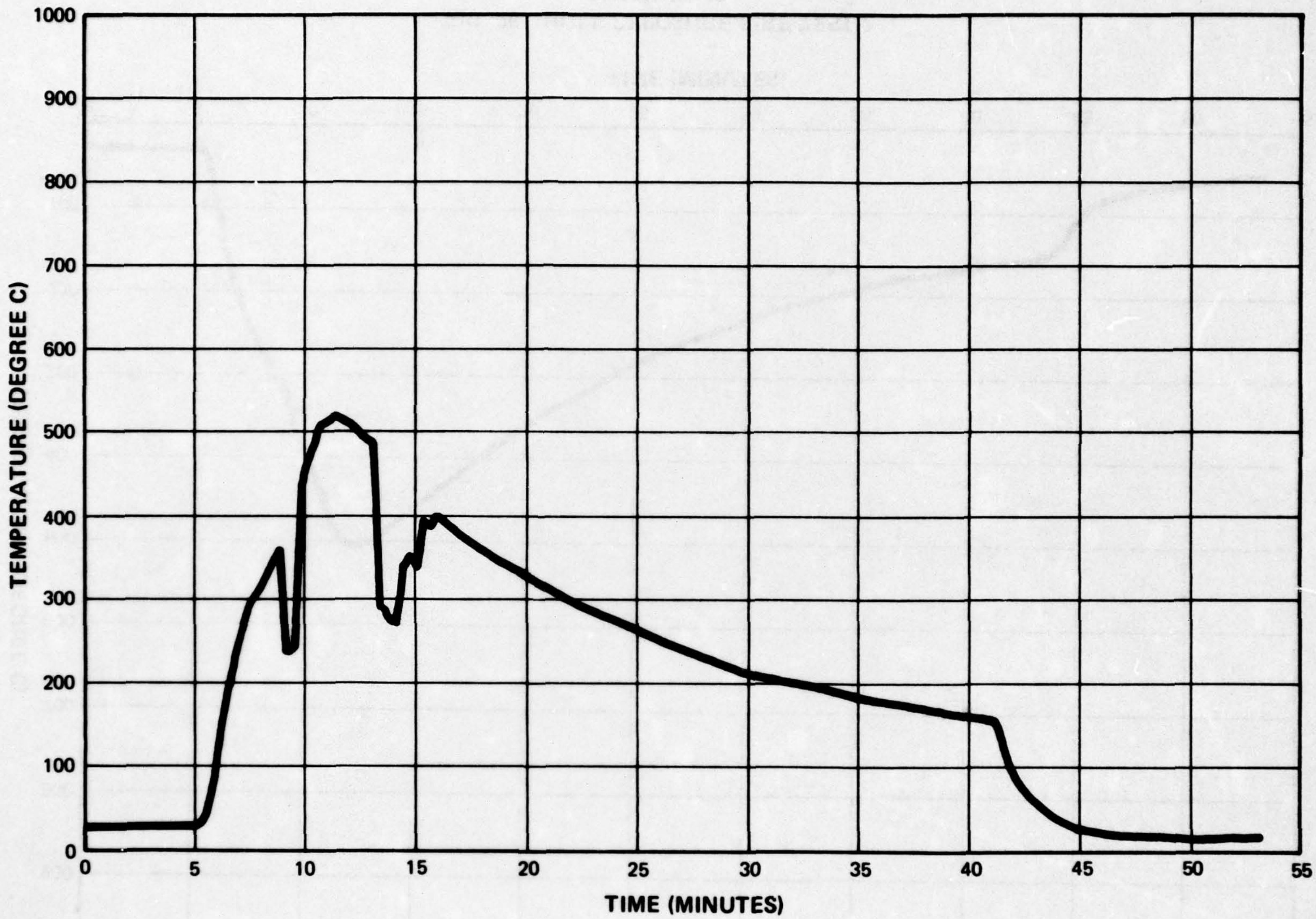


FIG. 24 HULL EXPOSURE FIRE TEST 2
CHANNEL 11

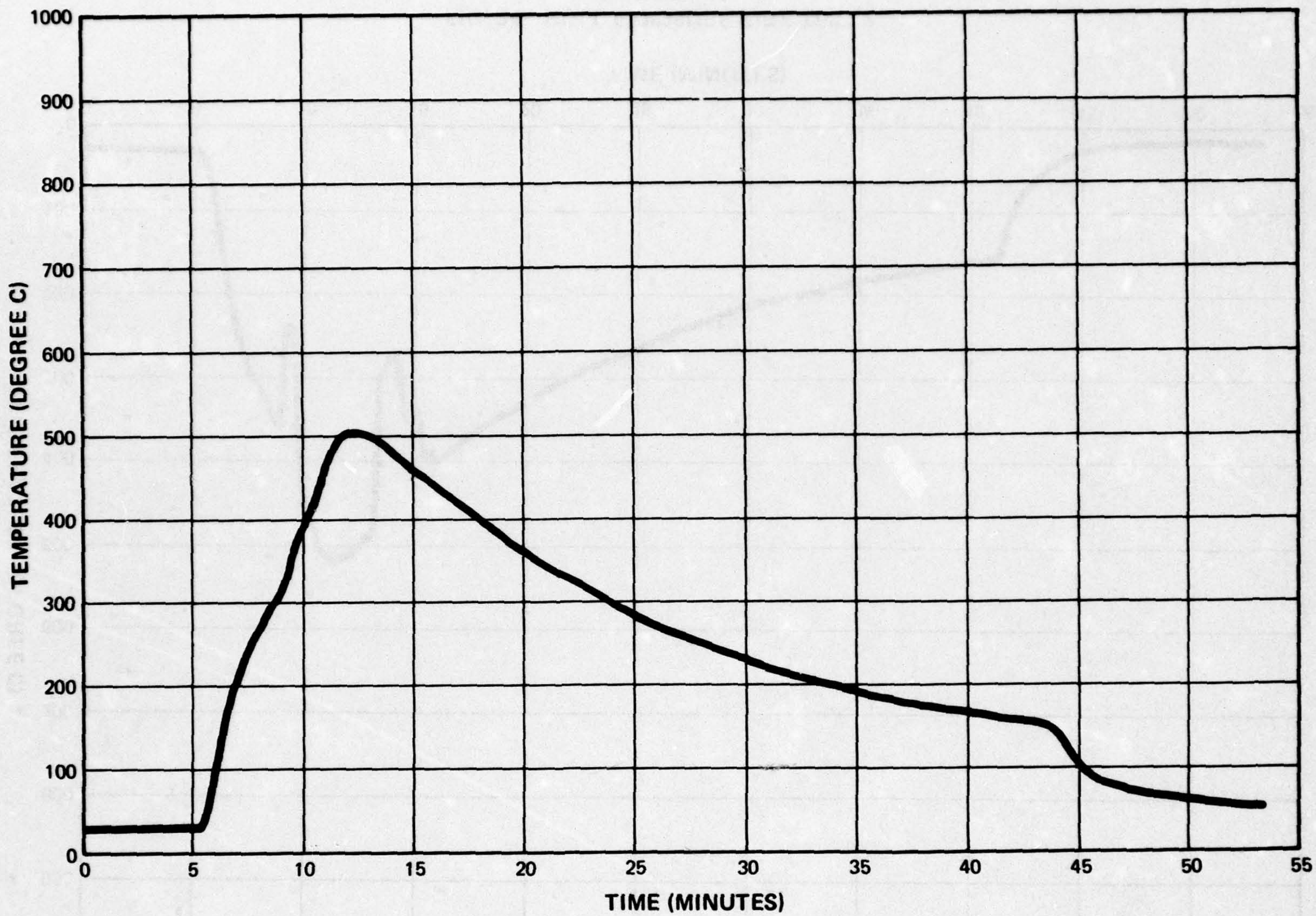


FIG. 25 HULL EXPOSURE FIRE TEST 2
CHANNEL 19

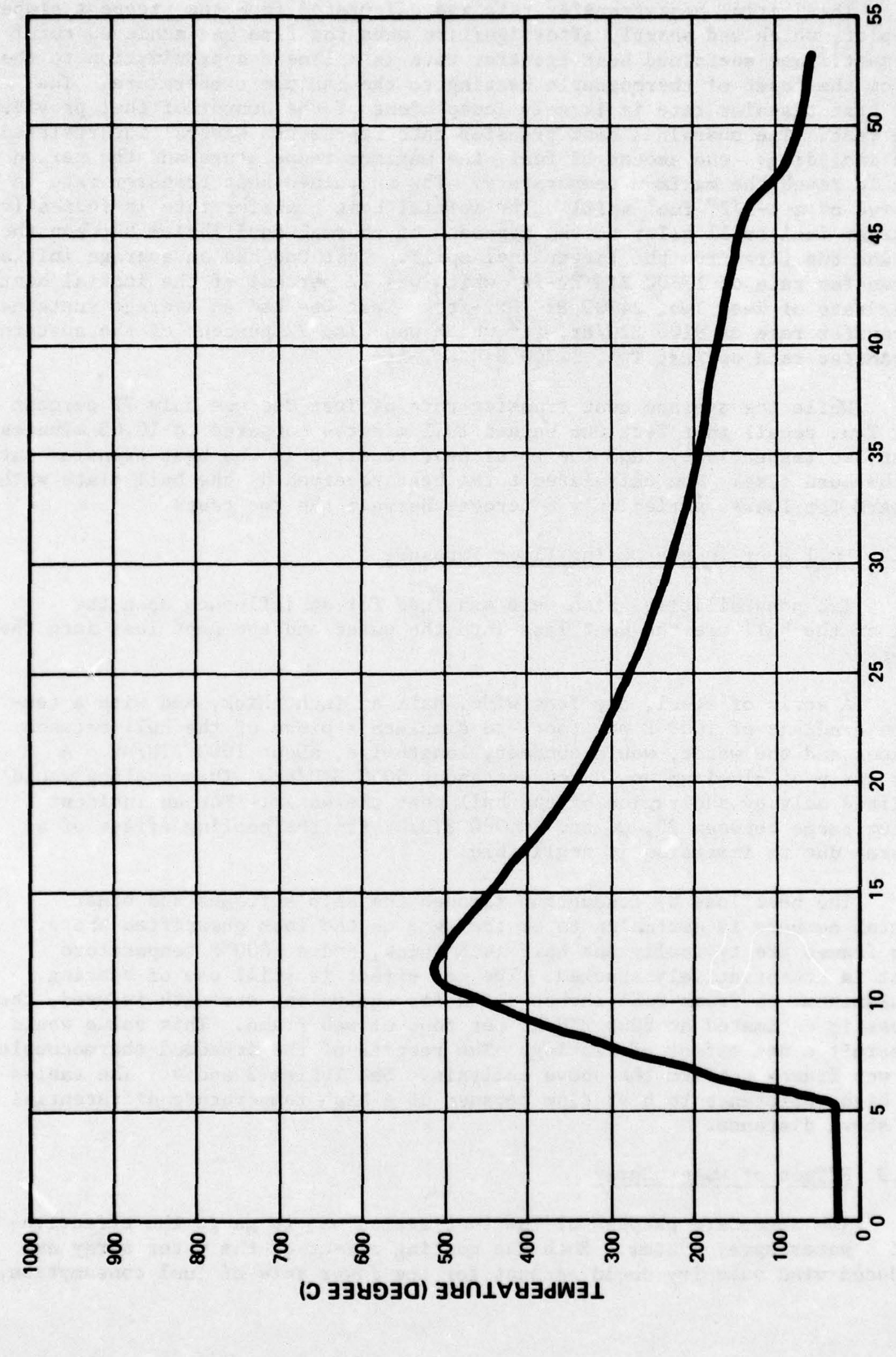


FIG. 26 HULL EXPOSURE FIRE TEST 2
CHANNEL 26

The initial heat transfer rate was calculated from the steepest slope of the plot, which was shortly after ignition when the fire had achieved total involvement. The sustained heat transfer rate is a linear approximation to the plot from the onset of thermocouple heating to the maximum temperature. The initial heat transfer rate is largely independent of the amount of fuel provided for the test. The sustained heat transfer rate represents several interrelated factors including: the amount of fuel, the maximum temperature and the period of time to reach the maximum temperature. The sustained heat transfer rate is indicative of a 1-1/2" fuel spill. The initial heat transfer rate is indicative of a larger fuel spill prior to the approach of thermal equilibrium between the vessel and the fire from the larger fuel spill. Test One had an average initial heat transfer rate of 17800 BTU/Hr-ft^2 which was 72 percent of the initial heat transfer rate of Test Two, 24700 BTU/Hr.-ft^2. Test One had an average sustained heat transfer rate of 9100 BTU/Hr.-ft^2 which was also 72 percent of the sustained heat transfer rate of Test Two, 12700 BTU/Hr.-ft^2.

While the average heat transfer rate of Test One was only 72 percent of Test Two, recall that Test One burned 13.1 minutes compared to 10.05 minutes for Test Two respectively, and the total heat received is the heat transfer rate times the burn time. The calculated total heat received by the hull plate without regard for losses varied only 6 percent between the two tests.

6.2 Hull Heat Losses During Flame Exposure

Two possibilities which were examined for an influence upon the cooling of the hull are the heat loss into the water and the heat loss into the structure.

A strip of steel, one foot wide, half an inch thick, and with a temperature gradient of 1000°F per foot, to simulate a piece of the hull between the flames and the water, would conduct, lengthwise, about 1000 BTU/hr. A similar strip of aluminum would conduct about 5000 BTU/hr. This cooling would be realized only by the region of the hull near the water. For an incident heat flux range between 20,000 and 40,000 BTU/hr ft^2 the cooling effect of a large area due to immersion is negligible.

The heat loss by conduction through the ship's frames and other structural members is estimated to be the same as the loss quantified above. The web frames are typically one half inch thick, and a 1000°F temperature gradient is conservatively assumed. The net effect is still one of heating. If an aluminum web frame 0.85 inches thick for equivalent strength is used, the heat loss is estimated at 8000 BTU/hr per foot of web frame. This value would still permit a net effect of heating. The results of the imbedded thermocouples in the web frames confirm the above analysis. See Tables 3 and 4. The tables show a high resistance to heat flow because of a high temperature differential over a short distance.

6.3 Effect of Water Spray

The secondary purpose of the test series was to gauge the effectiveness of a water spray system. Both the cooling effect of the water spray and the reduced wind velocity could account for the lower rate of fuel consumption,

TABLE 3

TEST 1

Peak Temperatures in Web Frames

Channel #	Location		Peak Temperature		@Time
	in	up	°C	°F	(min)
28	38	11	50°	122	35
29	24	11	80	176	30
30	12	11	150	302	30
31	6	11	545	1013	14
35	6	3	100	212	35
36	12	3	55	131	30
37	24	3	40	104	35
38	6	15	65	149	35
39	12	15	40	104	35
46	24	15	35	95	35

TABLE 4

TEST 2

Peak Temperatures in Web Frames

28	36	11	55	131	40
29	24	11	70	158	40
30	12	11	120	248	30
31	6	11	460	860	11
35	6	3	70	158	40
36	12	3	45	113	40
37	24	3	40	104	40
38	6	15	75	167	40
39	12	15	50	122	40
46	24	15	45	113	40

the lower air temperatures above the spray nozzles, the lower heat transfer rate and the lower average temperatures. The water spray could produce these results by local heat removal from the fire through a temperature rise and a phase change. While the decreased wind velocity could produce these results by a reduced transport of oxygen to the fire and to a lesser extent by a reduced fuel evaporation rate.

The wind velocity, though, was measured above the 02 deck level. The fire was on the lee side of the vessel and below the wind velocity sensor thus it is probable that the wind velocity effects on the two fires is less than 7 and 11 mph would incicate. In addition, the temperatures of Rows 1 and 2 are readily accounted for by the local cooling effect of the water spray but the data of Rows 3 and 4 correlate neither with the application of water spray or the reduced wind velocity.

Accordingly, the lower rate of fuel consumption, the lower air temperatures above the spray nozzles, the lower heat transfer rate and the lower average temperatures can be attributed primarily to the cooling effect of the water spray system.

The effectiveness of the water spray is exhibited by the temperatures reached by the top row of thermocouples. Refer to Figure 5. This top row is between the 02 and 01 decks and just below the water spray nozzles. No top row thermocouple exceeded 100°C with the water spray while without the water spray one top row thermocouple reached 632°C. The temperatures reached by the second, third, and fourth row thermocouples were considerably above 100°C and as high as 637°C in the second row with the water spray.

The direct effects of the water spray cooling can be seen in Figures 7, 8, and 9 by refering to Channels 1, 2, and 3. These channels had both fire involvement during both tests and protection from the water spray during Test One. The initial reduction in the heat transfer rate was 22000 to 25000 BTU/Hr ft^2 for all three channels based on an incident heat flux of 34000 to 37000 BTU/Hr ft^2.

Indirect effects of the water spray cooling can be seen in Row 2, Channels 9 to 16. Rows 3 and 4 show both increases and reductions in the heat transfer rate with no apparent correlation to the water spray of Test One.

6.4 Predicting the Heating of Aluminum

The tertiary purpose of this test was to predict the effect of a similar fire on an aluminum hulled vessel. If, as a result of a fire, melting of the hull or structural failure at elevated temperatures occurs, the extent of damage suffered by a vessel increases abruptly. Additionally, melting or structural failure of the hull would restrict the maximum survival time of persons aboard the vessel and present additional hazards to the cargo and the environment. Whether or not a part of the hull will melt depends on the heat transfer rate to the hull and the heat losses to the water or the inside of the ship. The most severe heating will take place when an aluminum hull is engulfed in flames from a flammable liquid floating on the water. A hull in direct contact with such flames will experience both radiative and convective modes of heating which can be assumed to result from a flame at a temperature of 1500°F.[5]

The heat transfer rate to the hull can be estimated from the results of the two tests. If the heat losses to the inside of the vessel and to the water can be quantified and found negligible then the mechanics of predicting the effect upon aluminum can be simplified.

Estimate of Plate Heat Retention

The overall heat transmission through the hull plate can be represented by:

$$U = \frac{1}{\frac{1}{h_o} + \frac{x}{K} + \frac{1}{h_i}}$$

where h_o = heat transfer coefficient due to radiative and convective heating from flame (BTU/hr-ft^2-F)

x = thickness of hull plate (ft)

k = thermal conductivity of plate (BTU/hr-ft/°F)

h_i = heat transfer coefficient due to natural convection to the inside air space

Values of 42.5 for h_o and 3 for h_i are within the range of commonly accepted values for surface coefficients under various conditions.(5),(6) For a 5/8 inch steel plate the overall heat transfer coefficient is assumed to be:

$$U = \frac{1}{\frac{1}{42.5} + \frac{.0625}{12.21} + \frac{1}{3}} = 2.8 \ \left(\frac{BTU}{hr\ ft^2\ °F}\right)$$

which is slightly less than the heat transfer coefficient for convection to the inside.

For the same plate, heat transfer into the plate and through the plate, but not to the inside of the ship is assumed to be:

$$U = \frac{1}{\frac{1}{42.5} + \frac{.0625}{12.21}} = 42 \ \left(\frac{BTU}{hr\ ft^2\ °F}\right)$$

and the hull plating retains 93 percent of the heat it receives. Thus, for thin plates of either steel or aluminum the hull plating will retain heat approximately in the ratio of h_o/h_o+h_i).

Note that this ratio is sensitive to parameters which are not known to exact values.

For the overall U into and through the plate to differ from h_o by 10 percent (10 percent is arbitrary) the thickness of the plate can be calculated. At this thickness, the effect of the temperature gradient across the plate thickness would no longer be considered negligible.

$$\frac{1}{h_o} = \frac{1}{42.5} = .024$$

For $\frac{x}{k}$ to be negligible

$$\frac{x}{k} < (10\%) \quad (0.024)$$

must be true.

$x < 0.0024\ k$ for $k = 31$, or

$x < 0.073$ ft or 0.9 inches of steel

For aluminum

$x < 0.0024\ k$ for $k = 118$, or

$x < 0.28$ ft or 3.3 inches of aluminum

Thus, for a h_o of 42.5, which results from direct contact with a flame, the temperature gradient in the steel plate to one inch thick and aluminum to 3 inches thick can be ignored, with negligible error. A reference to thin plate means less than one inch of steel or less than 3 inches of aluminum.

Verification of $h_o = 42.5$

By assuming no heat loss from the inside of the hull, the temperature of the flame can be determined.[5]

The equation was found to agree well with experiments for the penetration of aluminum aircraft by fire. This equation has the form of:

$$T = C + C_1 \exp [-\alpha t]$$

Rearranged to solve for flame temperature:

$$t_f = \frac{t_p - (t_p\text{-}1) \exp [h_o/5\rho c_p\sigma) \ (T\text{-}T\text{-}1)]}{1 = \exp [-(h_o/5\rho c_p\sigma) \ (T\text{-}T\text{-}1)]}$$

where t_f = flame temperature (°F)

$t_p\text{-}1$ = initial plate temperature at time T-1 (°F)

t_p = plate temperature at time T (°F)

h_o = total heat transfer coefficient due to the radiative and convective heating (BTU/lb °F)

ρ = density of plate material (lb/ft^3)

c_p = specific heat of plate material BTU/lb °F

σ = thickness of plate (in)

T = time of heating (min)

The constant of 5 in the equation would be unity if units of feet and hours were used instead of inches and minutes. The thermophysical properties in Table 5 were used in solving the equation.[4]

TABLE 5

THERMOPHYSICAL PROPERTIES

PROPERTY	STEEL	ALUMINUM	UNITS
k	31	118	BTU/hr ft °F
C_p	0.111	0.214	BTU/lbm °F
ρ	489	169	lbm/ft^3

Using the values of observed plate heating from the thermocouples, the flame temperature for Channels 19 and 20 for Tests 1 and 2 are presented in Figures 10, 11, 12, and 13.

Calculated flame temperatures reached peaks between 1400°F and 1500°F which are in good agreement with the recorded peak temperatures of the thermocouples in the air above the hull. Channel 43 of Test 2 reached a peak of 8500°C or 1562°F. Because the calculated flame temperatures agree with a commonly accepted 1500°F flame temperature and the measured values from the thermocouples in the air, the assumed value for h_o of 42.5 is reasonably accurate.

Effect on Aluminum

Once the flame temperature is known as a function of time, the effect of the same flame on a postulated aluminum hull can be predicted. For the predictions, two thicknesses of aluminum are plotted. One thickness plotted is the same thickness in inches as the steel plate. The other thickness plotted is an equivalent thickness of aluminum for equal strength at ambient temperature when compared to the steel plate. The equivalent thickness of aluminum is plotted as 1.7 times the thickness of steel. (This factor is dependent on the type of aluminum among other factors and should not be used for design purposes.) Also, this factor is representative of the strength ratio only at room temperature. When the aluminum is subjected to heating its strength decreases rapidly at comparatively moderate temperatures as shown in Figure 14 which compares the strength of aluminum with the strength of steel at elevated temperatures.

The flame temperature obtained from the heating of the steel hull is used in the following equation with the thermophysical properties of aluminum:

$$t_p = t_f - (t_f - t_p - 1) \exp [-h_o/5 \rho c_p \sigma) (T-T-1)]$$

The variables are the same as defined earlier. The result is the predicted heating of aluminum plate from exposure to the same fire.[5] The predictions for the heating of aluminum plate of equal thickness and equivalent thickness for equal ambient temperature strength in response to fires the same as the test fires can be found in Figures 10, 11, 12, and 13.

The plotted predictions show that aluminum of equal thickness heats 1.5 times as fast as steel. This factor comes from the time constant in the exponent of the equation:

$$\frac{\rho s\ Cps}{\rho al\ Cpal} = \frac{0.111 \cdot 489}{0.214 \cdot 169} = 1.5$$

If the aluminum thickness is changed for equivalent strength the heating factor becomes:

$$\frac{\rho s\ Cps\ \sigma s}{\rho al\ Cpal\ \sigma al} = \frac{0.111 \cdot 489}{0.214 \cdot 169\ 1.7} = 0.88$$

where $\frac{\sigma al}{\sigma s}$ is assumed to 1.7 for equivalent strength at room temperature.

Aluminum which is 1.7 times as thick as steel heats slightly slower than steel, if the temperature gradient in the plate and losses to the inside of the vessel are negligible. The heating of aluminum plate can also be predicted from a 1500°F flame instead of from a non-standard time temperature curve of an actual flame. For comparison purposes the actual heating of Channels 2 and 19 during Test 2 are plotted alongside the predicted heating of the steel plate from a 1500°F flame. Refer to Figures 15 and 16. Note that both the actual heating and the predicted heating exceed the requirements of a standard fire test which takes five minutes to reach 1000°F. The tow of the actual heating lags the predicted heating because the actual flame is still growing. In the next range the slopes are equal, which verifies the ability of the equation to predict plate heating. Differences at higher temperatures can be attributed to the randomness of a fire and the exhaustion of the fuel supply.

Using a 1500°F flame Figure 17 compares the predicted heating of very thin steel and aluminum plates while Figure 18 presents predicted plate heating of aluminum plates of thicknesses from 1/2 to 2 inches. These predictions for the heating of aluminum are presented mainly for information. Their use for design purposes is limited as unprotected aluminum of any practical thickness will melt if a 1500°F fire is of sufficient duration.

7.0 CONCLUSIONS

The tests were successful in that the data developed during the tests was able to provide, through analysis, a quantitative response to each of the three purposes of the tests.

Unfortunately, the one parameter with which many people feel comfortable, average peak temperatures, provides little information for valid comparisons between the two tests. See Table 1. While the higher temperatures of Test One could be explained by a longer burn time, air entrainment by the water spray or reduced soot because of the water spray and less black body absorption of heat, the water spray provides for increased heat absorption from the fire. Because the that received by the hull was nearly the same in each test (Test One was 6 percent less),

TEST	RELATIVE HEAT RECEIVED
1	72% heat transfer rate x 13.1 minutes = 9.4
2	100% heat transfer rate x 10.05 minutes = 10.0

Apparently, air entrainment, soot reduction, and heat absorption by the water spray had offsetting effects. As such, the higher peak temperatures of Test One are best explained by the extra burn time noting that both fires were fuel limited so that neither test approached thermal equilibrium.

Graphically, the heat received is roughly related to the area between the flame temperature and the steel temperature on Figures 10 through 13. If the flame temperature parallels the hull heating as it did in Test One, the hull reaches a higher temperature than the case in which the flame is initially very hot but cools as it continues to burn which occurred in Test Two.

The heat transfer rate for a vessel engulfed in an oil fire on the water was found to range from 35000 to 45000 BTU/Hr ft^2 for several channels of both tests. The abnormally high initial heating rate of Channel 45 probably means the hull plate is thinner than the 5/8" assumed.

The second purpose, gauging the effect of the water spray, did not lend itself to such ready quantification. Two identical fires would be the best basis for a comparison between the effects of water spray cooling and no cooling. In practice, however, two identical test fires can not be produced. As such, small overall variations between the tests and somewhat larger variations in a single channel between tests cannot be attributed to the water spray with much certainty. Variations between the tests when several channels are averaged can be attributed to the water spray with more certainty, assuming other variations between the tests, such as wind velocity, have been considered and dismissed. For example, with reference to Table 1 the maximum temperature of any channel, which is 637°C for Test 1 and 632°C for Test 2, is a single point and therefore should not be taken as indicative of the effects of water spray.

The 20°C reduction in the average of all maximum temperatures can, however, be attributed to the cooling effect of the water spray. But recall the entire test area did not benefit from the water spray. If only the top row is considered the average maximum temperature was 329° during Test 2 but only 82° during Test 1. This reduction of 247°C is best attributed to the cooling effect of the water spray.

The protected zone during the first test with the water spray was roughly the top one forth of the test area immediately below the spray nozzles. The other three fourths of the test area did not benefit directly from the water spray.

The cause of the small zone of protection was the placement of the spray nozzles at the top of the protected area with the expectation that a water film would protect the 35 foot high test area. Good design dictates that water spray nozzles be placed vertically at least every 12 feet.(3) The temperatures reached by row two thermocouples, 11 feet below those of row one reinforce the design criteria for regular vertical placement of nozzles.

Another item to consider is the effective rate of water application in the protected zone. The test was designed to protect with water spray an area 35 feet high by 40 feet in length for a total of 1440 square feet with 0.12 gal/min/ft^2. Even though the application rate is one half the normal rate when the total area is considered, the method of application restricted protection to the top row of thermocouples or 25 percent of the anticipated area of protection. The effective application rate of water over this reduced area was about 0.5 gal/min/ft^2. Even at an effective application rate of twice the normal design rate, only about 10 feet of the hull below the nozzles was effectively protected. This indicates placement of the nozzles at regular vertical intervals and uniform spray coverage is most important and compensation by even twice the recommended application rate of water is ineffective.

Because of the non-uniform rate of water application, no conclusion with respect to the effectiveness of water spray uniformly applied at 0.12 gal/min/ft^2 is possible.

The flame temperatures calculated for Channels 19 and 20 for Test One show the influence of the water spray. Refer to Figures 10, 11, 12, and 13. During Test One the flame temperature did not reach 1200°F until the fire had burned about five minutes. In Test Two the flame temperature reached 1200°F within two minutes. From the calculated flame temperatures, it appears that the water spray limited the initial rate of buildup of the fire. This limited rate of buildup delayed the burning of some of the fuel so that flame temperatures for Test One could exceed 1300°F nine minutes into the test. In contrast, the flame temperatures for Test Two did not reach 1300°F after the mid-point of the test.

The reduced maximum temperature reached under the influence of water spray is significant when fialure of the primary structure is the main concern.

The net heat input is significant from a cargo protection viewpoint.

From the maximum temperature readings, only the hull area which included top row thermocouples received protection from the water spray when protection of the primary structure is the main concern. The hull area which included both the top and second rows of thermocouples received protection from the water spray when net heat input to the cargo is the main concern.

It should be noted that water spray is primarily used to protect cargo because cargoes can be sensitive to net heat input regardless of the hull material. For this purpose, specific application rates have come into use. One should be cautious in transferring from cargo protection to primary structure protection via the common link of water spray without a thorough analysis of the validity of the carried over assumptions.

When a cargo is sensitive to heat input, the choice of water spray protection exhausts the practical protective options. Net heat input can be tolerated by some cargoes because relief valves can be sized for the expected net heat input. Vaporization of the cargo and its release completes the heat flow cycle from the fire thorugh the hull, into the cargo and back to the environment.

However, if the primary structure is sensitive to temperatures below the 1200 to 1500°F range, then net heat input is intolerable over a period of time. Net heat input from the exterior to a heat sensitive primary structure material is slightly more severe when the primary structure is insulated to provide protection from interior fires.

The insulation blocks the path for possible heat removal. This, however, is not a major concern, as discussed earlier.

The advantages of alternative materials are well known. The drawbacks include the following:

(1) A higher water application rate is necessary to reduce a low net heat input to a tolerable near zero net heat input to protect the primary structure.

(2) A water spray system, because it is an active system requiring proper design, periodic maintenance, a source of power, an activation mechanism, and the ability to function unattended throughout an emergency is not as reliable as steel, a passive fire protection system which sets a lofty standard.

(3) Casualties are often the result of a series of failures rather than a single incident and accordingly, the very incident that could have caused the fire could also have interfered with the operation of the water spray system, thus making the system useless for primary structure protection.(7)

(4) Failure of a water spray system for cargo protection would likely result in cargo loss but not necessarily loss of the vessel while failure of a water spray system for primary structural protection would likely result in loss of the vessel and its cargo as well.

(5) Aluminum fails by melting and provides no further protection while steel fails structurally by collapsing but it remains intact and can provide some further protection against flame impingement.

The result of a similar fire on aluminum would be disastrous.

The heating rate for aluminum is not much different from the heating rate of steel as explained earlier.

The test water spray system protected only the top 24 percent of the hull test area by virtue of a near zero net heat input. From a primary structure protection viewpoint, 75 percent of the hull test area was subject to net heat input.

While the test fires burned out just at the anticipated structural failure range or just short of the metling point of aluminum, a fire of longer duration would cause severe damage and loss of an aluminum vessel as a result of continued net heat input.

In view of the grave consequences for an aluminum vessel from close exposure to a floating flammable liquid fire which in accident situation are rarely

as brief as test fires, a risk analysis should be performed. It appears that a competent risk analysis will dictate the installation of a water spray system for some alternative primary structure materials.

Further, it is likely that the design water application rate cannot be reduced and still provide an effective water curtain for primary structure protection.

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